1 FUNDAMENTALS OF MACHINE DESIGN

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P. ORLOV

A technical and methodic handbook for the designers of machines, useful as a textbook for mechanical engineering faculties. Gives a systematic exposition of the rules for rational machine design on the basis of unification and standardization, and for building industrial machines incorporating in their design provisions for further development, lengthening of service life, and improvement of reliability. Surveys ways of reducing weight and increasing strength and rigidity. Considers problems of thermal stress and the effect of thermal deformation on the operation of parts and sub-assemblies. Describes methods of improving efficiency, increasing sturdiness, making assembly easier, and improving operating conditions.

FUNDAMENTALS OF MACHINE DESIGN

vol. 1

П. И. ОРЛОВ

ОСНОВЫ КОНСТРУИРОВАНИЯ

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«МАШИНОСТРОЕНИЕ»
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1 FUNDAMENTALS OF MACHINE DESIGN

P. ORLOV

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THE GREEK ALPHABET

A α Alpha	I ι Iota	P ρ Rho
B β Beta	K κ Kappa	Σ σ Sigma
Γ γ Gamma	Λ λ Lambda	T τ Tau
Δ δ Delta	M μ Mu	Υ υ Upsilon
E ε Epsilon	N ν Nu	Φ φ Phi
Z ζ Zeta	Ξ ξ Xi	X χ Chi
H η Eta	O o Omicron	Ψ ψ Psi
Θ θ Theta	Π π Pi	Ω ω Omega

This book has been recommended for republication by Indian specialists under the programme of the Joint Indo-Soviet Board to make the best Soviet textbooks available for Indian students.

На английском языке

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Preface

The purpose of the present book is to offer the reader an attempt at a systematic exposition of rules for rational designing.

With all the diversity of the modern machine-building the tasks facing the designer are similar in many respects. It is the reduction of the weight and specific metalwork weight of the machine, the improved suitability for industrial production, greater durability and reliability that are of importance for the design of any machine, the difference lying only in the relative significance of these factors. All this enables one to formulate the principles of rational designing as a code of general rules for machine building.

The prime intention of the book is to make the designer learn to work creatively. To design imaginatively means:

to abstain from blindly copying the existing prototypes and to design meaningfully, selecting from the entire store of the design solutions offered by the present-day mechanical engineering the ones that are most suitable under given conditions;

to be able to combine various solutions and find new, better ones, i. e., display initiative and put vim in the work;

to continually improve the machines' characteristics and to contribute to the progress in the given branch of mechanical engineering;

to follow the dynamic development of the industry and devise versatile machines of long life, amenable to further modernization and capable of meeting the ever-growing demands of the national economy without running the risks of obsolescence for a long time to come.

Particular attention in the book is attached to the problems of durability and reliability. The author endeavoured to strongly emphasize the role of the designer in tackling these problems.

In presenting the material the author followed the principle "qui vidit—bis legit" (the one who sees reads twice). Most of the designers are individuals of visual thinking and visual memory. For them a drawing or even a simple sketch means much more than many pages of explanatory notes. For this reason, each point in the text is accompanied by design examples.

To better the understanding most of the illustrations are arranged in such a way as to enable it to compare wrong and correct, inexpedient and expedient design versions.

The solutions given as correct are not the only possible ones. They should be regarded not as precepts, suitable for use in all cases, but rather as examples. In particular conditions other versions may prove more advisable.

Chapter 1

Principles of machine design

1.1. Objectives of Machine Design

The chief aim of the designer is to develop a machine that would satisfy most fully the needs of the national economy, would be most economic, and would have the best technical and operational characteristics.

The most important characteristics of machines are their productivity, efficiency, strength, reliability, weight, specific metalwork weight, size, power intensiveness, scope and cost of repairs, labour costs, service life, in-between repair times, degree of automation, simplicity and safety of maintenance, and convenience of operation, assembly and dismantling.

Any machine must meet the industrial design requirements, i.e., it must have a plain but attractive finish.

The priority of each of the above characteristics depends on the purpose of the given machine, namely:

for generators and energy converters the main characteristic is their efficiency which is indicative of the degree of useful energy conversion;

for power tools—productivity, precision and reliability of operation, and degree of automation;

for metal-cutting machines—productivity, accuracy of machining, and range of operations;

for control and measuring instruments—sensitivity, accuracy, and stability of readings;

for transport machines (particularly for air and spacecraft)—weight and engine efficiency which determine the amount of onboard fuel.

Economical considerations are of tremendous importance in engineering.

When designing a machine, the designer must do his best to make the machine as economical as possible throughout its service life.

This aim is achieved by way of enhancing the efficiency of the machine, increasing its service life, and cutting operational expenditures.

At the same time the designer must minimize laborious manufacturing operations, lower production costs, and reduce the time spent on designing, making, and running-in the machine.

A vast number of technological, organizational, processing, economic, and other factors affect the total cost of engineering products.

This book deals only with design methods which enhance efficiency and reduce production costs.

1.2. Economic Factors of Design

Economic factors must be made the basis of designing. Designing particulars should never overshadow the main aim—increase of machine efficiency.

Many designers consider that to design economically means cutting production costs, avoiding complex and expensive solutions, using cheapest materials and applying simplest processing methods.

This is but a part of the problem. The economic effect is determined by machine output and the total operational expenditures during service life. The cost of the machine is not the only and not always the main part of the expenditures.

Economy-oriented designing means consideration of all the factors determining the efficiency of the machine and a correct evaluation of the relative importance of each of these factors.

This principle is often ignored. In an attempt to obtain cheaper products the designer often achieves economy in one way only, while missing others and more effective ones. Moreover, such a one-sided economy, which disregards the totality of the essential factors, often results in a lower overall economy of the machine.

(*a*) *Profitability* (*Commercial Value*) *of Machine*

Machine *profitability* q is determined by the ratio between output (production) Ot over a certain period of time, expressed in terms of money, and the total operational expenditures Ex over the same period

$$q = \frac{Ot}{Ex} \tag{1.1}$$

The term "output" implies the cost of products made on the machine (the cost of finished and semi-finished products, and the cost of intermediate operations and useful work performed by the machine).

Generally, the total expenditures Ex cover the following: *De*—depreciation charges for the machine; *Pr*—cost of power consumed; *Mr*—cost of materials consumed; *Lbr*—cost of labour force; *Mntce*—cost of maintenance; *Ovhd*—overhead costs; *Rpr*—cost of repairs;

Gd—general depreciation charges for the plant, i.e.,

$$Ex = De + Pr + Mr + Lbr + Mntce + Ovhd + Rpr + Gd$$

The value of q must always be greater than unity, otherwise the machine will operate unprofitably, in other words, its existence will become commercially useless.

(*b*) *Economic Effect*

The *annual economic effect* (*annual profit*) Q from the machine is the difference between the annual output and expenditures

$$Q = Ot - Ex = Ot\left(1 - \frac{Ex}{Ot}\right) = Ot\left(1 - \frac{1}{q}\right) \tag{1.2}$$

where q is the profitability.

The *total profit* $\sum Q$ for the entire service life of the machine is equal to the difference between the total output $\sum Ot$ and total expenditures $\sum Ex$

$$\sum Q = \sum Ot - \sum Ex$$

or

$$\sum Q = \sum Ot - \sum(De + \sum Pr + \sum Mr + \sum Lbr + \sum Mntce + \\ + \sum Ovhd + \sum Rpr + \sum Gd) \tag{1.3}$$

The quantity $\sum Q$ depends on the duration of the machine operation. Let us introduce the following more precise definitions: *H*—*service life*, i.e., the total period (in years) of the machine's being in operation; *h*—*actual running time* (in years) for the entire service period. If we assume that the machine will run until its physical resources are fully exhausted, then, obviously, *h* is the *durability* of the machine, i.e., its potential running time.

The relation

$$\eta_{use} = \frac{h}{H} \tag{1.4}$$

is the *use factor* characterizing the operational intensity of the machine.

In Eq. (1.3) some terms ($\sum Rpr$, $\sum Gd$) are proportional to the *service life*, i.e., $\sum Rpr = HRpr$; $\sum Gd = HGd$, while the others ($\sum Ot$, $\sum Pr$, $\sum Lbr$, $\sum Mntce$, $\sum Mr$, $\sum Ovhd$), to the *actual running time* (i.e., to the machine durability, given the above assumption is valid) and are equal to hOt, hPr, etc., respectively.

The depreciation expenditures for the entire service life are equal to the cost of the machine

$$\sum De = C \tag{1.5}$$

Substituting the above values for the respective terms of Eq. (1.3) we will have

$$\sum Q = hOt - [C + h(Pr + Mr + Lbr + Mntce + Ovhd) + H(Rpr + Gd)]$$

Let us designate the expenditures proportional to the durability h as Ex' and those proportional to the service life H as Ex''. Then

$$\sum Q = hOt - (C + hEx' + HEx'') = hOt - \left[C + h\left(Ex' + \frac{H}{h}Ex''\right)\right]$$

Since, according to Eq. (1.4), $\frac{H}{h} = \frac{1}{\eta_{use}}$, then

$$\sum Q = h\left(Ot - Ex' - \frac{Ex''}{\eta_{use}}\right) - C \tag{1.6}$$

In terms of service life H the total profit (economic effect) is

$$\sum Q = H[\eta_{use}(Ot - Ex') - Ex''] - C \tag{1.7}$$

The *recoupment term T_r of the machine* is the service period for which the aggregate economic effect equals the cost of the machine ($\sum Q = C$). Substituting this expression in Eq. (1.7), we get

$$T_r = \frac{2C}{\eta_{use}(Ot - Ex') - Ex''} \tag{1.8}$$

When determining the recoupment term of the machine, the repair costs can be ignored because at the initial stages of operation they are insignificant.

(c) *Coefficient of Operational Expenditures*

The ratio between the total expenditures for the entire service period of the machine and its cost is called *coefficient of operational expenditures*:

$$k = \frac{\sum Ex}{C} = \frac{C + h\left(Ex' + \frac{Ex''}{\eta_{use}}\right)}{C} = 1 + \left(Ex' + \frac{Ex''}{\eta_{use}}\right) \tag{1.9}$$

Equation (1.6) can now be given in the following form

$$\sum Q = hOt - kC \tag{1.10}$$

The percentage ratio of the machine cost to the total expenditures is equal to the reciprocal of the coefficient of operational expenditures

$$c = \frac{C}{\sum Ex} = \frac{1}{k} \cdot 100\% \tag{1.11}$$

Coefficient k is usually much larger than unity and may be as great as 10-100.

As is seen from Eq. (1.9), the coefficient of operational expenditures increases with an increase in durability h of the machine. Correspondingly, the proportion of the machine cost in the total amount of expenditures decreases.

(*d*) *The Influence of Operational Factors upon Economic Effect*

Equation (1.6) shows that the overall economic effect, i.e., the total gain for the entire machine service life, is proportional to durability h. This gain will be the greater, the higher the annual output Ot and the less the machine cost C and expenditures Ex' and Ex''.

Let us consider the relative importance of each of these factors by analyzing the operation of an exemplary metal-cutting machine tool.

In this case it is best to determine the *net economic effect* $\sum Q'$ comprising the total profit less the cost of materials and consumable tools. Furthermore, we will ignore the general factory overhead which is difficult to consider and limit ourselves here to the overhead expenditures directly related to the operation of the machine (maintenance expenses are included in labour costs).

Let the machine cost C be 1500 roubles (rbl), power consumption of the machine drive electric motor, 10 kW. The machine operates on a double-shift basis with a load factor of 0.8. Taking into consideration holidays and Sundays (75 days per year), the machine use factor will be

$$\eta_{use} = 0.8 \cdot \frac{14}{24} \cdot \frac{365-75}{365} \approx 0.4$$

The actual machine running time per year will be

$$365 \cdot 24 \cdot 0.4 \approx 3500 \text{ h/year}$$

Assuming that on the average the machine operates at 0.75 of its rated power, the annual electrical power consumption is

$$0.75 \cdot 10 \cdot 3500 = 26\,250 \text{ kWh/year}$$

With an industrial tariff for power of 2.5 kopeks per 1 kWh, the annual cost of the power consumed is

$$Pr = 26\,250 \cdot 0.025 \approx 650 \text{ rbl/year}$$

If the annual operator's pay is 1500 rbl, then the cost of labour on a double-shift basis will be

$$Lbr = 2 \cdot 1500 = 3000 \text{ rbl/year}$$

Let the overhead rate be equal to 25% of the labour cost

$$Ovhd = 0.25 \cdot 3000 = 750 \text{ rbl/year}$$

Assume that the total cost of repairs by the end of the service life of the machine is equal to its cost, i.e.,

$$\sum Rpr = C$$

Then, the overall economic effect in terms of the service life is

$$\sum Q = H\,[Ot - (Pr + Lbr + Ovhd)] - \sum Rpr - C =$$
$$= H\,[Ot - (650 + 3000 + 750)] - 1500 - 1500 =$$
$$= H\,(Ot - 4400) - 3000 \qquad (1.12)$$

In order to calculate the output, assume that the profitability of the machine, related to the sum of expenditures *Pr*, *Lbr* and *Ovhd*, is

$$q = \frac{Ot}{Pr + Lbr + Ovhd} = 1.6$$

Then

$$Ot = 1.6\,(650 + 3000 + 750) \approx 7050 \text{ rbl/year}$$

and Eq. (1.12) becomes

$$\sum Q' = H\,(7050 - 4400) - 3000 = H2650 - 3000$$

On the basis of Eq. (1.12), let us analyze variations in the profit with an increase in the machine's service life and output, and also with changes in the cost of the machine, labour and power. Let the initial duration of service life H be equal to 2.5 years, which with the adopted use factor corresponds to the machine durability h of 1 year. The results of estimates for service lives of 2.5 to 25 years are given in Fig. 1 and Table 1.

From Table 1 and Fig. 1 the following conclusions can be drawn.

The profit sharply rises with the increase of h, i.e., with the increase of H, provided that $\eta_{use} = \text{const}$. Taking the profit for $H = 2.5$ years to be equal to unity, then with $H = 10$ years the profit increases by 6.5 times and with $H = 25$ years, by 17.5 times.

The coefficient of operational expenditures increases from its initial value of $k = 9$ up to $k = 73$ (with $H = 25$ years) with the increase in the machine's service life. This correspondingly lowers the ratio between the machine cost and the total operational expenditures. This ratio, equal to 11% with the initial service life figure (2.5 years), decreases to a negligible value (3-1.5%) when the service life is increased to more than 10 years.

Reduction of the machine cost appreciably influences the profit only when short service lives are involved. For example, reducing the cost in half (which is quite a sizable value) results in a 20.5-per

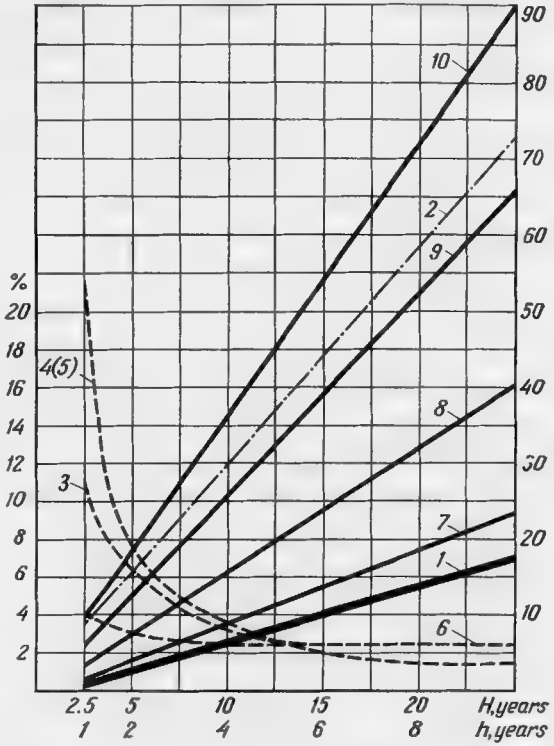


Fig. 1. Relation between overall economic effect and machine's service life H

1 — ratio $\sum Q / \sum Q_{z.s}$ for initial output and labour cost; *2* — coefficient of operational expenditures; *3* — ratio of machine cost to operational expenditures; *4* — increase of economic effect with machine cost halved; *5* — decrease of economic effect with machine cost increased 1.5 times; *6* — increase of economic effect with 10-percent increase in machine efficiency; *7* — ratio $\sum Q / \sum Q_{z.s}$ with labour cost reduced by 30%; *8* — ratio $\sum Q / \sum Q_{z.s}$ with output increased 1.5 times; *9*—ratio $\sum Q / \sum Q_{z.s}$ with output doubled; *10* — ratio $\sum Q / \sum Q_{z.s}$ with labour cost reduced by 30% and output doubled

cent increase in the profit when $H = 2.5$ years, but with a service life of over 10 years the increase comes to 3.5-1% only.

Conversely, the rise of the machine cost has a very slight effect upon the profit when the service life is long. Thus, increasing the machine cost by one half lessens the profit by 21% when $H = 2.5$ years and only by 3 to 1% with service lives longer than 10 years.

Table 1

Economic Effect as a Function of Service Life and Operational Factors

Economic indicators	Service life H, years					
	2.5	5	10	15	20	25
	Durability ($\eta_{use} = 0.4$), years					
	1	2	4	6	8	10
Profit $\sum Q$, rbl	3625	10 250	23 500	36 750	50 000	63 250
$\sum Q$ increase as compared with $\sum Q_{2.5}$	1	2.82	6.48	10.2	13.75	17.4
Coefficient of operational expenditures k	9	16.2	30.4	44.3	58.8	73
Ratio of machine cost to operational expenditures, %	11	6.15	3.3	2.25	1.7	1.4
Profit increase, %:						
with machine cost reduced by half	20.5	7.5	3.8	2	1.5	1.25
with machine efficiency increased by 10%	4	3	2.5	2.45	2.4	2.35
Profit decrease (in per cent) with machine cost increased by 1.5 times	21	7.3	3.2	2	1.5	1.2
Profit increase ($\sum Q / \sum Q_{2.5}$):						
with labour cost decreased by 30%	1.62	4.05	9	13.9	18.8	23.7
with output increased by:						
1.5 times	3.3	7.4	15.7	24	32.2	40.5
2 times	5.9	12.5	26	39.2	52.5	66
with output increased by 2 times and labour cost decreased by 30%	9.6	18	36.5	54.5	71.5	90

Consequently, making the machine more costly but of greater *durability* is quite justifiable economically since the gain from the enhanced durability by far exceeds the drop in the profit caused by the rise of the machine cost. Thus, increasing the initial durability by 6 times entailing even a two-fold rise of the cost increases the profit by $\frac{10.2}{1.04} \approx 10$ times.

Raising the efficiency of the machine (lowering the power costs) in our example has no significant effect. For instance, a 10-per cent increase in the efficiency raises the profit only by 4% when $H = 2.5$ years, and on the average by 2.5% when $H > 10$ years.

Profit is greatly increased by *lowering the labour costs* through automation, attendance of many machines by one operator, etc. Thus, decreasing expenditures on labour and maintenance and associated overhead charges by 30% enhances the profit by 9 times when $H = 10$ years and by 23.7 times when $H = 25$ years.

Increasing the machine output has a great effect. An output increase of 50% increases the profit by 15.7 times when $H = 10$ years and by 40.5 times when $H = 25$ years, these figures in the case of a doubled output being 26 and 66 times, respectively.

A sharp rise in the profit is obtained by *simultaneously increasing the durability and output of the machine and reducing the labour costs.* For instance, with the durability increased by 6 times, output doubled, and labour costs cut down by 30% the profit rises by 36.5 times when $H = 10$ years and by 90 times when $H = 25$ years.

The effect of the increase in the durability and output in this case is so great that it nullifies the influence of the other factors, e.g., the cost of the machine and power.

The above calculation example is sketchy since, apart from the assumptions that simplify computations, it does not consider the dynamics of operational changes (e.g., possible future alterations of the power cost, a fall in the output due to the gradual wear of the machine, etc.). Nevertheless, for machine tools the approximation clearly shows how operational expenditures influence the profit.

Naturally, for other machines, and with a different structure of operational expenditures, the influence of various factors on the profit will vary.

Let us take, for example, the cost of labour. There is an extensive range of machines (non-automatic machine tools; automobiles; road-building, constuction and agricultural machines, etc.), which cannot function without the constant attendance of an operator. Here the labour cost is comparatively high and there is no way to substantially reduce it. Correspondingly, as proved earlier, the relative importance of the machine cost within the totality of operational expenditures is not very great.

For machines that can run for long periods without any attendance (electric motors, electric generators, pumps, compressors, etc.) the labour costs only consist of the costs of periodic maintenance and inspection.

Most economic from the standpoint of labour costs are automatic and semiautomatic machines. For these machines the relative importance of the machine cost is much greater.

Power requirements of various machines differ considerably. For heat engines the fuel cost is of far greater importance than the cost of the machine and, occasionally, the cost of labour.

There are also machines which have an insignificant power consumption thanks to their high efficiency (electric generators, reduc-

tion gears, etc.). If, in addition, the labour costs are also small, the machine cost will then be the dominant factor.

All other things being equal, the machine cost to a decisive degree depends upon the number of the machines produced. When machines are produced on a mass scale their cost is not very high and its relative importance in the totality of operational expenditures is much slighter than in the case of machines produced on a small-lot basis or, the more so, to an individual order.

For some classes of machines the costs of depreciation, maintenance and repairs of industrial premises and installations carry great weight. These expenditures may by many times exceed those connected with running the machine.

An economic calculation similar to the above makes it possible in each particular case to determine the structure of operational costs and the relative importance of their constituents, and to establish an economically rational basis for designing the machine.

Generally, profit depends to the greatest degree on the *output and durability of the machine*. Therefore when designing a machine the designer must focus his attention on these factors. Equally important is the *reliability* which determines not only the durability but also the scope and cost of repairs carried out during the machine's being in use. In the previous example the consequence of the repair costs is somewhat overshadowed because in the calculation these costs for the service life of the machine have been taken at a moderate value equal to the machine cost. In other words, the repair costs are taken to be such as they must be for a correctly designed and reasonably run machine.

In practice repair costs may reach very large values exceeding in some cases the machine cost several times. Sometimes repair expenditures absorb a major part of the profit gained from operating the machine, thus making its further use unprofitable.

Nowadays the transfer of machines to repairless operation is a problem of precedence.

The term repairless operation implies:

exclusion of capital repairs;

exclusion of repairs to worn parts, meaning instead complete replacement by new parts, units and assemblies;

exclusion of emergency repairs, caused by the breaking or wearing down of parts, by way of planned maintenance.

The transfer of machines to repairless operation is a complex of problems the prerequisites for the Solution of which are as follows:

prolongation of the service life of wearing parts;

construction of machines on a unit assembly principle enabling independent replacement of worn parts and units;

provision in machines of non-wearing datum surfaces for locating changeable parts.

These design measures must be accompanied by technical and organizational measures, the most important of which is the organization of a centralized production of replacement parts and units.

The above said by no means implies that the designer may pay less attention to the problem of decreasing the machine cost. Such a conclusion would be wholly wrong. As noted earlier, the value of the machine cost depends on the type of the machine and may be great for machines having low energy and labour requirements, as well as for those with a comparatively short service life. It is only necessary to correctly evaluate the relative importance of this factor among other factors influencing the economy of the machine and be able to sacrifice it in cases when the reduction of the machine cost contradicts the requirements for enhanced output, durability and reliability.

It is noteworthy that along with decreasing the cost of individual machines, there is another, more effective way of lowering the cost of machine products as a whole, namely, reducing the list of machine products by selecting optimal types of machines, and satisfying the needs of the national economy while minimizing the number of the machine type-sizes (see p. 70).

Successful solution of the above listed problems should be the main activity of the designer, who must, first, set the policy in the machine building field and, second, develop progressive designs providing for high economic efficiency of machines, reduced operational expenditures and low cost of machine products.

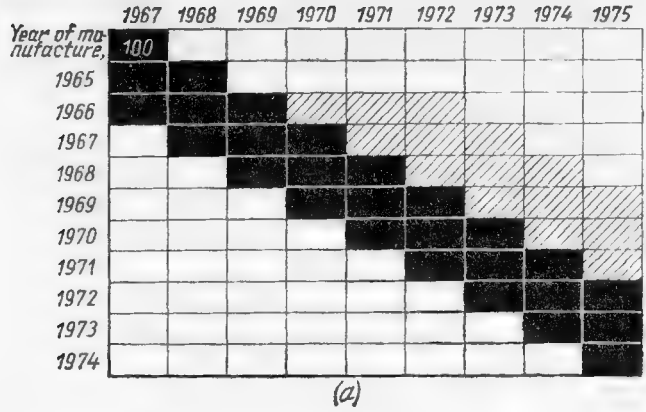
(e) *Influence of Durability on the Size of Machine Fleet*

Increasing the durability is an effective and economical means to increase the number of machines being used at one time.

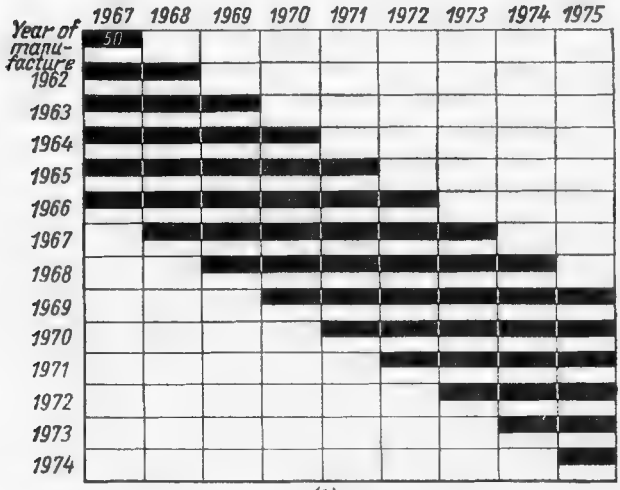
The number N of machines running at a given time is proportional to the product of their durability h and the number n of machines produced per year in the previous time.

As an example, consider a case when the annual production n is constant and equals 100 machines per year. Let the machine's durability h be 3 years; the machines operate continuously, i.e., their service life is equal to the durability.

The diagram in Fig. 2*a* pictures the utilization of the machines by the years. The number of the machines manufactured yearly is shown by blackened rectangles. The sum of the rectangles along the horizontal shows the duration of the machines' being in service, equal in this case to three years; the sum of the rectangles along the vertical represents the number of the groups of machines produced in different years and being in operation at one time. This quantity is equal numerically to the durability ($h = 3$), provided the annual production of the machines and their durability remain constant.



(a)



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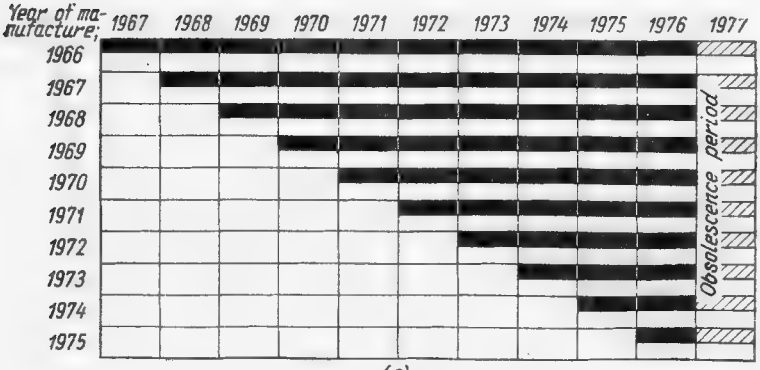


(c)

Fig. 2. Machine utilization diagrams

(a) with $\eta_{use} = 1$; (b) with $\eta_{use} = 0.5$; (c) with $\eta_{use} = 0.3$

Hence, the total number of the machines in use every year is

$$N = hn = 3 \cdot 100 = 300$$

Now assume that starting from 1966 the manufacturer doubles the durability (hatched rectangles). The machine's service duration, represented by the horizontal sum of the blackened and hatched rectangles, becomes $h' = 6$ years. From 1970 onward the size of the machine fleet increases, reaching in 1972 a stable value which for all the subsequent years remains constant and equal to 600 machines, i.e., to the product of the new durability and the annual production

$$N' = h'n$$

Thus, with the production of the machines being the same, a two-fold increase in the durability raises by as many times the size of the machine fleet and, cosequently, the annual output of products (and the total output for the entire service period of the machines as well).

Let us examine a case when a machine is not fully used, i.e., when the machine's service period is lengthened in comparison with its durability reserve (Fig. 2*b*).

The service life H is equal to the quotient of durability h by the use factor η_{use} accounting for all forms of forced or scheduled stoppages

$$H = \frac{h}{\eta_{use}}$$

Let $h = 3$ as before, and $\eta_{use} = 0.5$, then the service life $H = \frac{3}{0.5} = 6$ years.

The reduction of the degree of utilization of machines is equivalent to reducing the number of the machines operating at one time. In our case ($\eta_{use} = 0.5$) this reduction is shown on the diagram by halving the height of the blackened rectangles. The number of the machines of the same year's manufacture, operating simultaneously in the course of one year becomes equal to 50.

The number of the groups of machines of different years' manufacture, running at the same time is equal to the vertical sum of the rectangles. With the initial assumptions ($n = \text{const}$, $h = \text{const}$) being valid, this quantity for any year is numerically equal to the machine's service life ($H = 6$).

The total number N of machines in the fleet for any year is the product of the service life and the actual number of the machines of each group operating at the same time ($n_{act} = n\eta_{use}$)

$$N = Hn\eta_{use}$$

But

$$H = \frac{h}{\eta_{use}}$$

Hence

$$N = hn$$

In the case being considered $N = 3 \cdot 100 = 300$.

The result can easily be checked by simply finding the total number of machines in any one of the vertical columns of the diagram ($N = H \cdot 50 = 6 \cdot 50 = 300$).

Thus, the annual size of the machine fleet in operation does not depend on the use factor or the service life, but only on the durability and the number of machines manufactured yearly.

This conclusion, naturally, holds true only if the total service life of the machine is equal to its mechanical life. However, when the service life is limited because of obsolescence the picture sharply changes: the machine is rendered obsolete before its mechanical life has expired and one has to prematurely discard it, thus losing products which could have been manufactured if the durability reserve of the machine had been fully utilized during a shorter period of time.

Let us illustrate this (Fig. 2*c*). Let the machine's durability be 6 years. Assume that, due to a low shift factor, prolonged downtimes, etc., the machine utilization at any given moment is equal to 30%. With the service life corresponding to the full utilization of the durability reserve, i.e., with $H = \frac{6}{0.3} = 20$ years, the machine (as is obvious from the above) will yield production equal to $6A$, where A is the annual output. With an annual manufacture of machines $n = 100$, the total output of the machines of one year's manufacture will be $600A$, which for 10 years (1966-1975) becomes $6000A$.

Now assume that the obsolescence period is 10 years. Then, in 1976, i.e., 10 years after their manufacture, the machines must be discarded. Up to this time the 1966 machines will yield only 50% of their potential production ($0.5 \cdot 600A = 300A$); 1967 machines 45%; 1968 machines, 40%, etc. The total output of all the machines manufactured in 1966-1975 will be $1650A$, i.e., $\frac{1650}{6000} \cdot 100\% = 27.5\%$ of the production they would have yielded if their durability reserve had been fully utilized. Thus, the limits imposed because of obsolescence sharply reduce the total output. In our example the manufacture of machines, which in the near future would become obsolete, would inevitably lead to huge losses.

The analysed cases belong to the simplest ones. The picture becomes much more complicated with annual changes in the number

of the machines manufactured and their durability. Yet, the general regularity remains valid: increasing the durability (within the limits of the obsolescence period) is always accompanied, in the years to come, by an increase in the actual size of the machine fleet, the increase being proportional to the size of the annual manufacture of the machines and their durability.

(*f*) *Influence of Durability on Output*

Let us now examine the problem of the output obtained from a group of machines working simultaneously.

The total output (in terms of money) obtained from a machine throughout its entire service life H is equal to the product of the annual output Ot and actual running time of the machine

$$\sum S = Ot \cdot H \cdot \eta_{use}$$

Assuming that the machine fully exhausts its durability reserve ($H\eta_{use} = h$),

$$\sum S = Ot \cdot h \tag{1.13}$$

The annual output from a group of machines will equal the product of the annual output from each machine, use factor η_{use}, and number N of the machines being in operation at one time

$$\sum S_{annl} = Ot \cdot \eta_{use} \cdot N \tag{1.14}$$

The number N of machines running in any given period of time is equal to the product of the number n of machines produced yearly and their service life H

$$N = nH$$

Substituting this relation into Eq. (1.13), we obtain

$$\sum S_{annl} = Ot \cdot n \cdot H \cdot \eta_{use} = Ot \cdot n \cdot h \tag{1.15}$$

Hence, the total production delivered by a machine throughout its service life [Eq. (1.13)] and the annual output from a group of simultaneously running machines [(Eq. (1.15)] are proportional to the product of the annual output and the durability of the machine.

Doubled durability will double the annual production. If, simultaneously, the output is also doubled, the total output will then be increased four times. If the annual output is specified, then the increase of the durability and output will allow the number of the machines produced yearly to be reduced in proportion to the product nh. In this instance the total expenditures on the machine manufacture and labour will also be reduced, which means added financial gain.

Using the numerical values from the above example, we may compute the increase in financial gain resulting from the reduction of the number of machines manufactured yearly made on account of increased durability and output of the machines.

Assume that the required annual production is provided for when the number of machines manufactured yearly is 100, the machines having service life $H = 5$ years and annual output $Ot = 7050$ rbl/year.

With the doubled service life and output ($H = 10$ years, $Ot = 14\ 100$ rbl) the number of machines to be made each year to provide for the specified production will be reduced to 25 machines.

According to Eq. (1.12), the total output of the machine for the entire service life

$$\sum Ot = H \cdot Ot$$

The total gain from 100 machines with service life $H = 5$ years and annual output $Ot = 7050$ rbl is

$$\sum Ot_{100} = 100 \cdot 500 \cdot 7050 = 3\,525\,000 \text{ rbl}$$

The expenditures over the entire service life of the machine [Eq. (1.12)] are

$$\sum Ex = H\,(Pr + Lbr + Ovhd) + \sum Rpr + C$$

Substituting the numerical values from the above example ($Pr = 650$ rbl/year, $Lbr = 3000$ rbl/year, $Ovhd = 750$ rbl/year, and $\Sigma Rpr = C = 1500$ rbl), we obtain

$$\sum Ex = 5\,(650 + 3000 + 750) + 1500 + 1500 = 25\,000 \text{ rbl}$$

The total expenditures for 100 machines

$$\sum Ex_{100} = 100 \cdot 25\,000 = 2\,500\,000 \text{ rbl}$$

The total profit from 100 machines

$$\sum Q_{100} = \sum Ot_{100} - \sum Ex_{100} = 3\,525\,000 - 2\,500\,000 = 1\,025\,000 \text{ rbl}$$

Assume that the cost of machines with service life $H' = 10$ years and output $Ot' = 14\ 100$ rbl/year is equal to the doubled cost of the original machines ($C' = 2C = 3000$ rbl). Consider also that the difficulty of repairs increases proportionally to the machine cost, that is, $\sum Rpr' = 2C = 3000$ rbl.

Then, the total expenditures over the entire service life of the machine

$$\sum Ex' = H'\,(Pr + Lbr + Evhd) + \sum Rpr' + C' =$$
$$= 10\,(650 + 3000 + 750) + 3000 + 3000 = 50\,000 \text{ rbl}$$

The total expenditures for 25 machines

$$\sum Ex'_{25} = 25 \cdot 50\,000 = 1\,250\,000 \text{ rbl}$$

The total output from 25 machines possessing the higher durability and output will, according to the original equation, be the same as for 100 machines

$$\sum Ot'_{25} = 25 \cdot 10 \cdot 14\,100 = 3\,525\,000 \text{ rbl}$$

The total gain from 25 machines will be

$$\sum Q'_{25} = \sum Ot'_{25} - \sum Ex'_{25} = 3\,525\,000 - 1\,250\,000 = 2\,275\,000 \text{ rbl}$$

Consequently, when replacing 100 machines with 25 machines of higher durability and output, the profits rise by $\frac{2\,750\,000}{1\,025\,000} = 2.2$ times in spite of the fact that the machine cost is doubled.

The picture will only slightly differ, if the power consumption is taken to be proportional to the output, i.e., when $Pr' = 2Pr = 1300$ rbl/year. In this case

$$\sum Ex' = 10\,(1300 + 3000 + 750) + 6000 = 56\,500 \text{ rbl}$$

For 25 machines

$$\sum Ex'_{25} = 25 \cdot 56\,500 = 1\,412\,500 \text{ rbl}$$

The total financial gain

$$\sum Q'_{25} = 3\,525\,000 - 1\,412\,500 = 2\,112\,500 \text{ rbl}$$

i.e., compared with the original estimated profit for 100 machines, the gain rises by $\frac{2\,112\,500}{1\,025\,000} = 2.05$ times.

Conclusions. The increase of the machine's output and durability is a most effective and profitable means to raise the production and financial gain.

The durability increase allows a proportional reduction in the number of the machines produced annually to be made without decreasing the total industrial output, and thus provides for cutting down the cost of the machines manufacture, materially reducing the operational expenditures, and raising the overall economic effect.

On the other hand, increasing the machine's durability without any changes in the annual number of the machines produced, their output and the cost of their manufacture will ensure a larger machine fleet, raising accordingly the total industrial output.

It is obvious that enhancing the machine's durability as a means of increasing the size of the machine fleet, their total output, and the power intensiveness of the national economy is by far more

profitable than simply increasing the number of the machines without raising their durability.

Increased machines manufacture is achieved by commissioning new plants, expanding the production areas and equipment of the existing enterprises, or by increasing the productivity of the existing equipment through intensification of production processes (which is most advisable economically).

In the first and second cases subject to increase are the costs of the machines manufacture, while the operational expenditures rise in all of the cases because of the increased number of the machines in operation.

As a rule, an increase in the output and durability of machines is only accompanied by a relatively small rise of their cost, and at the same time it leads to a reduction in the operational expenditures. The final result is the same: a larger actual size of the machine fleet and greater overall industrial output, but with incomparably lesser expenditures and materially greater overall economic effect.

Thus, we may conclude that in mechanical engineering it is a good policy to combine the increase of the machines manufacture with the increase of their output and durability and, in certain cases, to moderate the machines manufacture, giving preference to the more advantageous way of increasing their output and durability.

Also, from the above we may conclude that increasing the annual machines manufacture does not necessarily mean an increase in the number of actually working machines and the total industrial output. The figures of a yearly rise of the machines manufacture have no value as an economic growth indicator if they are not supplemented with objective data on the durability of the machines and the quality of production they yield.

These figures may mean: progress, if the durability remains at a constant level or increases; stagnation, if the durability lowers in the same proportion as the output grows; and regression, if the purability falls to a greater proportion than does the output grow.

(*g*) *Output*

Output is expressed by the cost of products or of effective work done by the machine in unit time. The output depends on the machine's productivity, i.e., on the number of operations (or work units) performed in unit time, and also on the cost of operations (work units).

To increase the output is a complex problem, the solution of which depends to a large extent upon the correct operation of the machine. For motor vehicles, for example, operational means of increasing the output include the elimination of empty runs, aug-

mentation of running speeds, increase of the load-carrying capacity (use of trailers), etc. The productivity of machine tools is improved by intensifying the technological procedures and by using special attachments and fixtures.

But in the main this problem must be solved at the design stage. The machine must be made of the highest possible productivity according with the actual requirements of the production and prospects of its development. The business end of the machine should be designed with a view to maximizing the scope of operations by appropriately selecting the kinematic scheme, power capacity, strength and rigidity.

The main ways to enhance the productivity of machines are as follows:

increasing the number of simultaneous operations performed on the work;

increasing the number of workpieces being simultaneously machined;

automating the production process.

The first method is most fully expressed in the design of multihead metal-cutting machines enabling workpieces to be machined on several or all surfaces simultaneously. An automatic multiple-tool lathe is another example.

The second trend is represented by rotary machines on which several components are processed at the same time. Another example is a batch-type processing, in which several workpieces in multiseat fixtures are worked upon at a time.

Rotary machines are, in fact, roundabout-type machines provided with rotating tables. Several workpieces which are to be subjected to one or several operations are mounted on the rotary table and allotted to an individual operative member (block, chuck or spindle), also fixed on the table, and are fully machined as the table makes one complete revolution.

Rotary machines possess a high productivity (e.g., rotary filling machines with 40-60 spindles perform up to 1500 filling operations per minute).

The high productivity of rotary machines is due to their continuous operation and the simultaneous processing of many components. The number of components processed simultaneously is

$$z = \alpha i$$

where i = number of operative units positioned along the periphery of the rotary table;

α = circumference portion over which the processing operation takes place

The productivity of a rotary machine

$$Q = i \cdot n \text{ parts/min}$$

where n = rotary table speed, rpm;
i = number of operative units positioned on the rotary table

The speed of the rotary table is dependent upon the time during which the operation is being performed (so-called phase time)

$$n = \frac{\alpha}{t_{phase}}$$

where t_{phase} = phase time, min;
α = circumference portion over which the processing operation takes place (generally $\alpha = 0.65\text{-}0.75$)

Consequently,

$$Q = i \frac{\alpha}{t_{phase}} \text{ parts/min}$$

From this equation it is obvious that the productivity of a rotary machine can be increased by:

decreasing the phase time (i.e., speeding up the operation);
increasing the number of operative units.

The possibilities of decreasing the phase time are limited, whereas increasing the number of operative units enables the productivity to be raised, practically, as much as desired. The limiting factor in the latter case is the speed of loading and unloading the workpieces.

Rotary machines can be easily arranged to form automatic lines offering high productivity, compactness and economic effect (thanks to the reduction of the amount of power consumed in transferring workpieces from one machine to another).

1.3. Durability

Similarly to the output the durability of a machine is heavily dependent upon the conditions and technical level of its use: care of the machine, timely maintenance, and prevention of overloads—all these measures may greatly add to the machine's durability. Conversely, low-skilled, poor attendance shortens the machine's service life.

But it is the correct design of the machine that determines its durability.

(*a*) *Durability Criteria*

The durability of a machine is the total time during which the machine can work under normal conditions and still retain its principal ratings, due account being taken of repairs, provided their total cost is within reasonable limits.

Occasionally, application also finds the notion of durability reserve, i.e., the time in hours of machine running until the first major overhaul.

In many cases, particularly for machines of aperiodic action, durability is measured in terms of the total output for the entire service life. In this case the durability is the total number of operations (units of work) that the machine or unit is capable of performing until complete wear out. For instance, the durability of motor vehicles and railway rolling stock is determined as the maximum distance travelled; that of instruments and testers, as the total number of switchings; of foundry mills, as the total number of melts; and of soil-cultivating machines, as the total area cultivated.

Depending on operating conditions, the actual durability may substantially differ from the rated one. It falls due to systematic overloads, speeds higher than rated, increased working forces, adverse climatic conditions, etc. Conversely, the durability increases when the machine operates under lighter conditions.

The influence of operating conditions upon the durability may be taken account of by introducing a duty factor η_{duty}. Thus, the actual durability will be

$$h' = \frac{h_{rated}}{\eta_{duty}} \tag{1.16}$$

where h_{rated} is the rated durability.

The magnitude of this factor can be reliably determined on the basis of a differentiated investigation into the influence of operating conditions and duty of the machine upon its durability. This is a problem pertaining to the statistical theory of durability. If precise data is not available, one may take the following approximate values: for light duty — $\eta_{duty} = 0.7\text{-}0.8$; medium duty — $\eta_{duty} = 1$; heavy duty — $\eta_{duty} = 1.2\text{-}1.5$.

(*b*) *Service Life*

The service life of a machine is the total period it remains in operation up to complete exhaustion of its durability reserve.

For machines of aperiodic action the service life is determined as the quotient of durability h, expressed in terms of the number of operations (work units), by the average number of such operations per year.

For instance, the service life of an automobile rated for a total run of L kilometres is

$$H = \frac{L}{\eta_{duty} l} \text{ years}$$

where η_{duty} = duty factor;

l = average distance (in km) travelled by the automobile per year

When dealing with machines whose durability is expressed in units of time the total service life is the quotient of the durability h by the *use factor* η_{use}, the latter showing the average degree of actual machine utilization throughout the entire period of its service. Considering also the duty factor

$$H = \frac{h}{\eta_{duty}\eta_{use}} \tag{1.17}$$

In a general case the use factor is

$$\eta_{use} = \eta_{seas} \cdot \eta_{off\text{-}d} \cdot \eta_{rep} \cdot \eta_{shift} \cdot \eta_{mach} \cdot \eta_{load} \cdot \eta_{acc.\ br} \tag{1.18}$$

The *seasonal work factor* η_{seas} is the ratio between the duration h_{seas} of the working season of the machine (in days) and the total number of days in the year

$$\eta_{seas} = \frac{h_{seas}}{365}$$

Included in the category of machines whose work is dependent upon climatic and seasonal conditions are most of agricultural machines (combine harvesters, cultivators, croppers, etc.), as well as the machines used for processing perishable agricultural products (equipment for canneries), road-building machines, snow-clearing vehicles and river transport vessels which have a limited period of navigation.

For special-purpose agricultural machines, e.g., potato diggers, cotton pickers, etc., $\eta_{seas} = 0.05\text{-}0.02$.

For machines used the whole year round $\eta_{seas} = 1$.

The *off-day factor* $\eta_{off\text{-}d}$ is the ratio of the number of working days to the total number of days in the year (365). Off-days include Sundays (52 days per year) and national holidays (on the average 7 days per year). Besides, the off-day factor takes account of the shortening of the working days preceding the national holidays, which on the average, amounts to 4% of the total annual working time.

Thus, the off-day factor averages to

$$\eta_{off\text{-}d} = \frac{365-52-7}{365} \cdot 0.96 \approx 0.8$$

This factor is valid for machinery working to a calendar schedule. For machines run continuously throughout the year (e.g., equipment of power stations) $\eta_{off\text{-}d} = 1$.

The *shift factor* η_{shift} is the ratio of the duration h_{shift} of the working shifts of the machine in hours to the 24-hour day

$$\eta_{shift} = \frac{h_{shift}}{24}$$

For single-shift operation $\eta_{shift} \approx 0.3$; for double-shift operation $\eta_{shift} \approx 0.6$; for triple-shift operation $\eta_{shift} \approx 0.9$; and for 24-hour operation $\eta_{shift} = 1$.

The *repair downtime factor* η_{rep} is the ratio between the duration h_{act} of the actual operation of the machine and the sum of the actual operation duration h_{act} and repair downtime duration h_{rep}

$$\eta_{rep} = \frac{h_{act}}{h_{act} + h_{rep}}$$

The magnitude of this factor depends primarily on the machine's reliability, which defines the length of time between repairs and the scope of the latter, and on the level of organization of repair work.

Factor η_{rep} depends also on the time of the machine's being in use. Being quite negligible initially, repair downtimes lengthen progressively as the machine wears down and may reach by the end of the service life quite a great value.

For processing machines operating to a calendar schedule the average value of η_{rep} equals 0.85-0.95. For other machine categories this factor varies very widely.

For machines of seasonal and sharply aperiodical operation η_{rep} is equal to unity because such machines are almost always repaired during their off periods.

The *machining time factor* η_{mach} is determined as the ratio of the machining time h_{mach} (actual time of machining) to the sum of machining time h_{mach} and auxiliary time h_{aux} (time spent in mounting and removing the work, setting up, adjusting and servicing the machine)

$$\eta_{mach} = \frac{h_{mach}}{h_{mach} + h_{aux}}$$

This factor is valid for manually-controlled machine tools (e.g., metal-cutting machines, metal-forming equipment). The vaule of η_{mach} depends on the type of equipment, degree of perfection of the working process, work batch size. For a metal-cutting machine tool this factor on the average equals 0.8-0.9. As the degree of automation goes up η_{mach} approaches unity. For automatic transfer machines η_{mach} equals unity.

The *load factor* η_{load} is the ratio of the duration h_{act} of the actual operation to the sum of the duration of the actual operation and the duration h_{dt} of the downtimes for the same period, the downtimes being due to the impossibility of loading the machine fully

$$\eta_{load} = \frac{h_{act}}{h_{act} + h_{dt}}$$

For machines running without any hourly schedule, as well as machines of periodic action whose loading is not regulated (auxiliary, emergency, repair), η_{load} is low.

Special-purpose machines may be underloaded in a continuous production process when they perform a limited number of operations on a narrow component range. The same is true of machines whose output capacities exceed those of the neighbouring machinery.

As a rule, a low value of factor η_{load} in processing machines is due to planning defects and incorrect choice of equipment as far as its quantity, type and capacity are concerned.

In production processes with changeable objects of production the magnitude of this factor depends upon the component being machined and upon the type of machining, and hence may vary with time. For example, if turning operations are predominant, then the lathes will be the most occupied machines, whereas machines of other types (milling, boring, etc.) will either be underloaded or idle.

Under the conditions of small-batch production the average value of η_{load} equals 0.7-0.75, in batch production—0.8-0.85, aud in multiple production—0.9-0.95. In stable continuous large-lot production $\eta_{load} = 1$.

The *accidental breakdown factor* $\eta_{acc \cdot br}$ is the ratio of the duration h_{act} of the actual operation to the sum of the duration h_{act} and idle time $h_{acc.br}$ caused by breakdowns eliminated on the spot

$$\eta_{acc.\ br} = \frac{h_{act}}{h_{act} + h_{acc.\ br}}$$

For well designed and carefully operated machines this factor is close to unity. For machines having design defects, or for those operated by unskilled personnel, $\eta_{acc.\ br.}$ may be much lower than unity.

Let us prove Equation (1.18).

In a generalized form the use factor is

$$\eta_{use} = \frac{h_{act}}{F}$$

where h_{act} = total number of machine's running hours per year;
$F = 365 \cdot 24 = 8760 =$ annual time fund, hours

Hence,

$$\eta_{use} = \frac{8760 - \sum h_{dt}}{8760} = 1 - \frac{\sum h_{dt}}{8760} \qquad (1.19)$$

where $\sum h_{dt}$ is the total number of downtime hours per year.

Downtime for non-working seasons

$$h_{seas} = 8760\,(1 - \eta_{seas})$$

Downtime for off-days

$$h_{off\text{-}d} = 8760 \eta_{seas}\,(1 - \eta_{off\text{-}d})$$

Downtime for repairs

$$h_{rep} = 8760\eta_{seas}\eta_{off\text{-}d}(1-\eta_{rep})$$

Downtime for the non-working part of the day (24 hours)

$$h_{shift} = 8760\eta_{seas}\eta_{off\text{-}d}\eta_{rep}(1-\eta_{shift})$$

Downtime for the non-machining period

$$h_{mach} = 8760\eta_{seas}\eta_{off\text{-}d}\eta_{rep}\eta_{shift}(1-\eta_{mach})$$

Downtime because of incomplete machine loading

$$h_{load} = 8760\eta_{seas}\eta_{off\text{-}d}\eta_{rep}\eta_{shift}\eta_{mach}(1-\eta_{load})$$

Downtime because of operating troubles

$$h_{acc.br} = 8760\eta_{seas}\eta_{off\text{-}d}\eta_{rep}\eta_{shift}\eta_{mach}\eta_{load}(1-\eta_{acc.br})$$

Summing up all the above downtimes gives

$$\sum h_{dt} = 8760\,(1-\eta_{seas}\eta_{off\text{-}d}\eta_{rep}\eta_{shift}\eta_{mach}\eta_{load}\eta_{acc.br})$$

Substituting this expression in Equation (1.19) we obtain

$$\eta_{use} = 1 - \frac{\sum h_{dt}}{8760} = \eta_{seas}\eta_{off\text{-}d}\eta_{rep}\eta_{shift}\eta_{mach}\eta_{load}\eta_{acc.br}$$

(c) *Design Durability. Design Service Life*

For general-purpose machines running to a yearly time-schedule with preset repair downtimes the degree of utilization, and hence the difference between service life H and durability h, will depend mostly on the shift factor η_{shift}.

Figure 3 shows the relation between H and h for different conditions of work. The diagram is plotted under the following assumptions: $\eta_{duty} = 1$ [Eq. (1.16)]; $\eta_{seas} = 1$; $\eta_{off\text{-}d} = 0.81$ (except for all-the-year-round continuous operation when $\eta_{off\text{-}d} = 1$); $\eta_{mach}\eta_{load}\eta_{acc.\,br}\eta_{rep} = 0.8$.

On these assumption the service life is

$$H = \frac{h}{0.81 \cdot 0.8\eta_{shift}} = \frac{h}{0.648\eta_{shift}}$$

where $\eta_{shift} = 0.3;\ 0.6;\ 0.9$ (for single-, double- and triple-shift operation, respectively).

For all-the-year-round continuous operation

$$H = \frac{h}{0.95}$$

where factor $\frac{1}{0.95}$ is introduced to account for accidental and repair downtimes.

Considering the graph, one may conclude that the durability reserve of a machine, sought for at the design stage, must accord with the use factor η_{use} and, in the first place, with the shift factor η_{shift}. Raising the durability of machines which would be used not so very intensively will result in an increase in their service

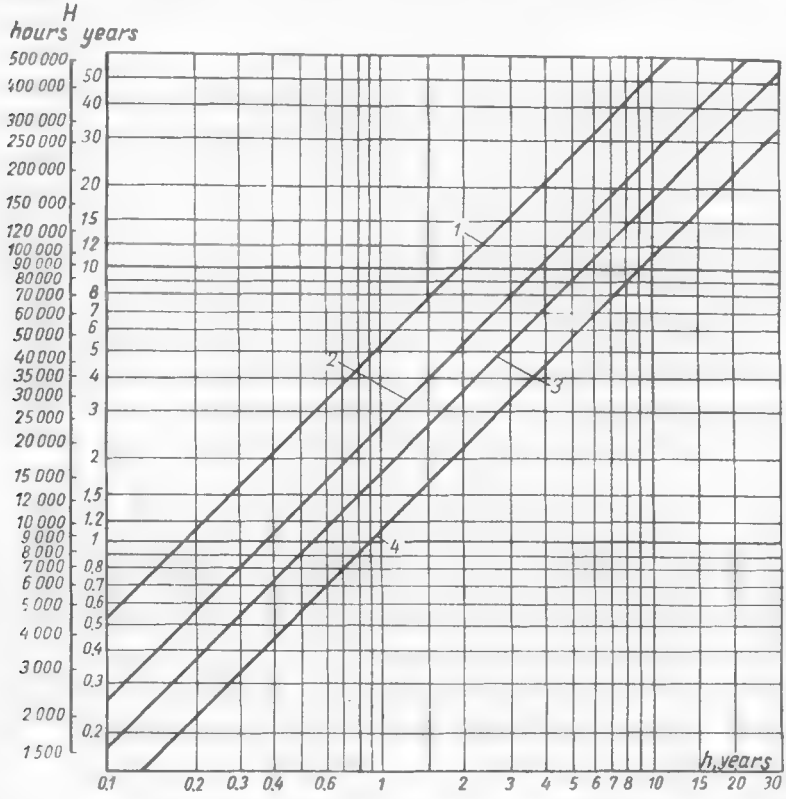


Fig. 3. Durability h versus service life H

1 — single-shift operation; *2* — double-shift operation; *3* — triple-shift operation; *4* — all-the-year-round operation

life which cannot be practically used to the full because of obsolescence. For instance, when durability $h = 10$ years, the service life under the conditions of a double-shift work will be 28 years, and under those of a single-shift operation, 50 years, which surpasses all conceivable obsolescence limits.

It is advisable to use high design durability values for machines which would be intensively operated. Thus, machines with a rated 10-year durability, operating to a triple-shift schedule have their service life equal to 17 years, and in the event of a continuous yearly

operation, to 11 years, a value which for most machine categories falls within obsolescence limits.

Table 2 gives rounded-off values of design durability (determined on the basis of the above diagram) for machines of different operative intensities, with the machines' service lives being preset. This data can be used for determining approximately the design durability of machines of different classes.

Table 2

Design Durability as a Function of Service Life and Operative Intensity

Service life, years	Design durability, thous. hours			
	single-shift operation	double-shift operation	triple-shift operation	continuous (yearly) operation
1	1.8	3.5	5.2	8
2	3.5	7	10	16
3	5.2	10	16	24
5	9	18	27	40
10	18	35	55	80
15	27	55	80	120
20	35	70	105	160
25	45	90	135	200

In a most common case of a double-shift operation of machines with a service life of 10-15 years the durability rating of the machines ranges from 40 000 to 60 000 hours. These figures may be taken as the basis for determining the durability of most of processing machines. For intensively operated machines (i.e., for those operating to a triple-shift schedule or continuously all the year round) the durability values should be taken at 60 000 to 100 000 hours, the machines' service lives being the same.

(*d*) *Theory of Durability*

The theory of durability is now being developed. Its main subjects are as follows:

determination of technologically and economically reasonable limits of durability;

development of methods for studying the operation of machines (statistical processing of actual operational data);

study of operating conditions and their influence upon the machines' durability; standardization of ranges of operating conditions;

determination of the degree of utilization of machines, and correlation between the machines' durability and service life;

diagnosis of the causes of failures;

revelation of parts limiting the durability; study of the effect of the components' durability upon the durability of the machines as a whole;

development of methods of bench and field tests of machines, units and elements for durability; prediction of the machine's durability on the basis of the bench test data;

development of reliable durability indices for machines being produced.

When defining the durability, the multiplicity and heterogeneity of factors influencing it (technological level of operation, fluctuations

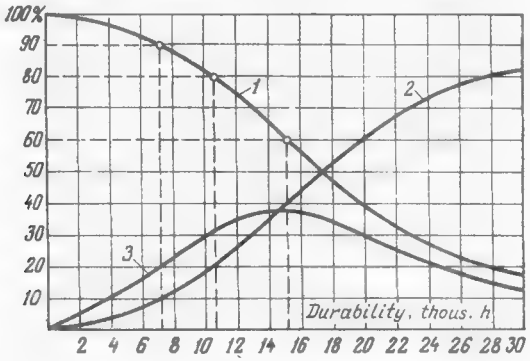


Fig. 4. Graph of probable durability

1 — probable service life (survival percentage); *2* — probability of destruction; *3* — density of service life probability

of operating conditions, quality of manufacture, etc.) and indeterminacy of many other factors (scattering of the strength characteristics of materials, effects of different regional and climatic conditions, etc.) necessitate the use of the probability theory and mathematical statistics. Because of this, the theory cannot provide an unambiguous solution for the problem of anticipated durability, restricting itself to the determination of a functional relation between the probability of failure and the duration and conditions of operation (Fig. 4). The theory can only define that probable duration of the machine's operation under given conditions will be, say, 7200, 10 500 and 15 000 hours at a failure probability of 90, 80 and 60%, respectively, or define probable number of machines remaining in operation (survival percentage) after certain periods of work.

Further, it is necessary to take account of the type and degree of failures, i.e., to determine with a certain confidence whether vital

or less important parts or units have failed, whether the machine remains repairable, what are probable scope and cost of repairs.

From these positions the durability of a machine may be defined as probable duration of the machine's operation under specified conditions, with which the probability of the machine's failure does not exceed a certain preset limit (say, 10%), the machine remains repairable, and probable cost of repairs does not exceed a definite value expressed, say, as a percentage of the machine cost.

The formulation of durability standards is a difficult problem and requires the collection and processing of vast information.

The study of machine durability would become much easier if every new machine were equipped with a "workmeter", i.e., with an hour-counter or operation-counter (similar to the odometers on automobiles). All new machines should be fitted with such devices.

Conclusions drawn from the studies of actually operating machines refer to machines manufactured in the past years and are thus inevitably out-of-date. As a result, such conclusions in essence are inapplicable to new machines incorporating latest design and technological improvements. Therefore predictions of the durability of new machines, which are of vital practical importance, have to be based upon bench tests of the machines (or new units incorporated into them).

Thus, one of the chief aims of the theory of durability is the development of accelerated test methods and correlation of such tests with actual operation.

The theory of durability, based on statistical data, is essentially applicable to articles of mass production and in a much lesser degree, to small-lot, and still less, individual products. Generally speaking, it should be noted that the theory of durability, as formulated earlier, is based on phenomenological grounds and operates with figures of the already achieved durability. Of much greater importance is the development of methods for enhancing the durability. Here the study of the physical regularities of failure, wear and damage of machine components (depending on the type of loading, properties of materials, condition of surfaces, etc.) comes to the fore. The problems listed above are so differentiated and specific that it is hardly possible to confine them within the framework of a general theory of durability. Such problems are solved by the methods of the theories of strength and wear, and particularly through concentrating the activities of designers and production engineers upon the enhancement of the durability of machines.

(*e*) *Means to Enhance Durability*

The main factors limiting the durability and reliability of machines are as follows: breakage of parts; wear of friction surfaces; surface damages caused by contact stresses, work hardening and

corrosion; plastic deformations occurring because of local or general stresses exceeding the yield limit, or creep (at high temperatures).

Strength in most cases is not an insurmountable limit. In general-purpose machines breakage can be fully excluded. With the available range of modern machine-building materials, the existing methods of manufacture, and the present state of the theory of strength, the machines of this class have no parts that cannot be made of virtually limitless durability.

The problem is more difficult in the case of highly-stressed machines, such as, for example, transport vehicles. Size and weight requirements make it necessary to increase the design stresses, which results in a greater probability of failure. Nevertheless, the continuous improvement of strengthening technology and refinement of computation methods make it possible, even in this case, to eleminate or significantly widen the strength limits of durability.

Many chance factors can be minimized: manufacturing factors (fluctuations in the mechanical properties of materials, technological defects)—through careful product quality control; operation factors (overloads, wrong handling of machines)—by purely design methods (by introducing protective systems, safety devices, interlocks, etc.).

Heat engines are in the worst position. Their durability depends primarily on the endurance of parts operating under high temperatures (pistons, piston rings, and valves in internal combustion engines; rotor and stator blades in steam and gas turbines; combustion chambers in gas turbines).

The strength of materials sharply drops with an increase in temperature. Furthermore, high temperatures give rise to the creep phenomenon (i.e., plastic flow of material under the effect of comparatively low stresses) which results in a change in the dimensions of parts, and hence, in the loss of their efficiency.

Machine parts operating under high temperatures have limited durability ratings. The service life of such parts can only be increased by design methods (reducing stresses, proper cooling) and, mainly, by using heat-resistant materials (high-alloyed chrome-molybdenum, chrome-vanadium-molybdenum, chrome-tungsten-molybdenym steels, titanium alloys, nickel-base alloys). For manufacturing thermally stressed parts nowadays wide use is made of sintered metal-ceramic materials (cermets). Cermet materials are made with oxides, nitrides and borides of Ti, Cr and Al, and carbides and nitrides of B and Si as the base, the bonding material being metallic nickel, cobalt and molybdenum.

Practically, the durability of machines is mostly dependent on the wear of their components. Gradually developing wear worsens the machine characteristics, lowers the accuracy of the operations done, and results in a poorer efficiency of the machine, greater

energy consumption and lower profits. Wear may eventually become ruinous. The progressive destruction of surfaces causes failures (breakage of antifriction bearings, pitting of gear teeth, etc.).

The main kind of wear in machines is mechanical wear which is subdivided into abrasive, sliding friction, rolling friction, and contact wear. Some parts are subject to chemical wear (corrosion), thermal, and cavitation-erosion wear. The multiplicity of kinds

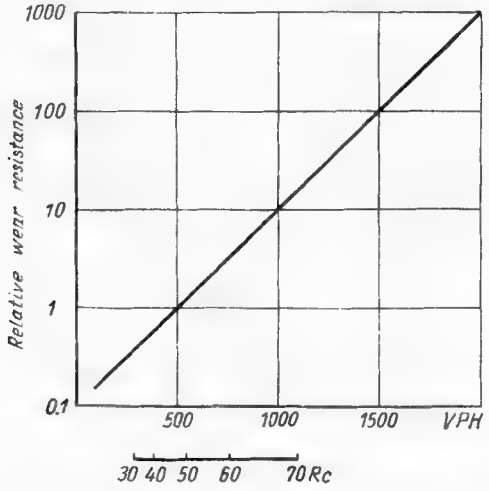


Fig. 5. Wear resistance *versus* surface hardness (after Goodwin)

of wear and the differences in their physical-mechanical nature require a differentiated studies into wear and special methods for its prevention.

The main methods for improving the resistance to mechanical wear are as follows: increasing the hardness of rubbing surfaces, selecting properly the material for friction pairs, decreasing the unit pressure on friction surfaces, improving surface finish, and ensuring correct lubrication.

Figure 5 shows the effect of surface hardness on the resistance to wear, the graph being plotted on the basis of experiments on the wear of surfaces under the action of an abrasive material (corundum).

The wear resistance of a surface with a Vickers pyramid hardness (VPH) of 500 ($\approx$ 48 Rc) is taken as the measurement unit. As evident from the diagram, each 500-VPH increment of the surface hardness gives a 10-fold increase in the resistance to wear.

The experimental conditions (abrasion wear) differ from the actual conditions of operation of lubricated surfaces in machines.

Nevertheless, these experiments give an idea of the huge influence that the surface hardness has on the resistance to wear.

Modern technology has at its disposal effective methods for raising the surface hardness: case-hardening and induction-hardening (500-600 VPH), nitriding (800-1200 VPH), beryllizing (1000-1200 VPH), diffusion chromizing (1200-1400 VPH), plasma carbide facing (1400-1600 VPH), boronizing (1500-1800 VPH), boron-cyaniding (1800-2000 VPH).

Another trend is to improve the antifriction properties of surfaces by depositing phosphate films (phosphating), by saturating the surface layer with sulphur (sulphidizing), graphite (graphitization), molybdenum disulphide, etc. Though having a moderate hardness, such surfaces exhibit a greater slipperiness, low friction coefficient, and high resistance to tearing, jamming and seizure.

The above methods (especially sulphidizing and saturation with molybdenum disulphide) increase the wear resistance of steel parts by 10-20 times. Combined methods also find application. This can be exemplified by the method of sulphocyaniding which simultaneously enhances the hardness and slipperiness of surfaces.

Of vital importance is the correct combination of the hardnesses of friction surfaces. Maximum hardness of both surfaces in contact is advisable when they operate at low relative speeds and with high loads imposed thereon, whereas operation at high relative speeds and in the presence of a lubricant requires a combination of a hard and a soft surfaces, the latter preferably possessing better antifriction properties.

An efficient way to improve the wear resistance of friction joints is to decrease the unit pressure therein. Occasionally this can be achieved by reducing the loads (rational distribution of forces) or by minimizing the cyclic and impact character of loads. But most simply this can be effected by increasing the area of the friction surfaces, often accomplished without any appreciable increase in the overall dimensions of machines.

To illustrate, let us examine a machine tool guideway acted upon by a unilateral load (Fig. 6*a*). An appropriate change of the guideway profile (Fig. 6*b*) makes it possible to double the bearing surface area, halve the unit pressure and, hence, improve the durability, with the overall guideway dimensions remaining the same. Ridged guideways offer still greater durability (Fig. 6*c*). In this case the unit pressure is reduced four times, the original overall dimensions being only doubled.

If the design permits, it is strongly recommended that point contact be changed to line contact, line contact, to surface contact, and sliding friction, to rolling friction. Point contact gears are to be avoided; these include transmissions with skew axes, spiral bevel gears, helical wheels with large spiral angles, and screw gears.

The latter have a further disadvantage as their contact spot moves rapidly along the tooth under sliding friction conditions, while in conventional involute gears rolling friction occurs at a rather low speed.

A special method is wear compensation accomplished either automatically or periodically. Units with a periodic wear compensation include sliding (plain) bearings in which radial or axial clearances, can be adjusted (bearings with tapered trunnions or seating surfaces, or with inserts or bushes that can be periodically tightened up).

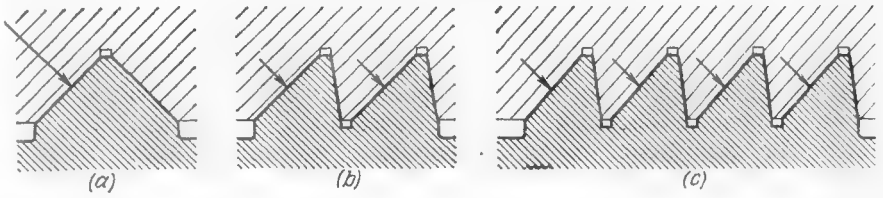


Fig. 6. Reduction of unit pressure on friction surfaces (case of machine-tool guideways)

Another example of periodic wear compensation is the axial tightening of antifriction bearings (radial thrust or tapered) or adjustment of clearances in rectilinear guides by means of tightening wedges and wear strips.

More perfect are systems with an automatic wear compensation (self-grinding-in conical plugs of taps, face- and lip-type seals, spring-preloaded antifriction bearing units, hydraulic clearance compensation systems in lever-type mechanisms, etc.).

Proper lubrication of friction units is of decisive importance. Wherever possible, fluid friction should be ensured, thus eliminating semifluid and semidry friction.

Open mechanisms lubricated by way of a periodic packing should be avoided. Open gear transmissions must never be used, and chain drives should better be avoided.

All rubbing parts should be enclosed in housings and reliably protected against the ingress of dust, dirt and atmospheric moisture.

The best solution is to use hermetically sealed systems provided with a continuous forced oil feed to all the lubrication points.

Excessive lubrication is not recommended for units operating in conditions of high periodic contact loads and speeds (antifriction bearings, gear teeth, etc.). It is advisable to lubricate such units with a metered stream of lubricant, while at higher rotational speeds atomized (oil-mist) lubrication is preferable.

The viscosity and thermal characteristics of oils must be compatible with the operating conditions of the given unit or machine.

The efficiency of lubricants can be enhanced considerably by adding to them special substances which improve their lubricity (colloidal graphite and sulphur, molybdenum disulphide) and oiliness (oleic acid, palmitic acid, and some other organic acids), preclude oxidation (organic and metal-organic compounds containing sulphur, phosphorus, and nitrogen) and prevent seizing (organosilicon compounds).

When the application of liquid oils is either impossible (operation under high temperatures, in chemically aggressive media, in high vacuum) or inefficient (under the conditions of high-frequency contact loads) use is made of dry-film lubricants: graphite, molybdenum disulphide (MoS_2), lead monoxide (PbO), cadmium oxide (CdO), lead iodide (PbI_2), cadmium iodide (CdI_2), lead sulphide (PbS), etc. Dry-film lubricants are usually employed in the form of films applied to metal surfaces. To improve their lubricity and endurance properties, such films are reinforced with binders (powdered metallic nickel, silver, and gold).

An ideal alternative, from the standpoint of wear resistance, is complete elimination of any metal-to-metal contact between the working surfaces. Examples of wear-free units are electromagnetic clutches and brakes in which torque is produced on account of electromagnetic forces developing in the gap between their working surfaces.

Approximating, in principle, to a wear-free operation are sliding bearings with a hydrodynamic lubrication. This is achieved thanks to a continuous delivery of oil and provision of a wedge-shaped oil gap making for the delivery of the oil into the loaded zone; under steady operating conditions the metallic surfaces in such bearings are fully separated, thus making, theoretically, the bearing unit wear-free. The vulnerable point in this case is the disturbance of fluid friction in non-stationary conditions, particularly during starts and stops, when the delivery of oil is discontinued due to the reduced rotational speeds, causing a metal-to-metal contact to develop between the trunnion and the bearing and thereby intensifying their wear.

Recently use has been made of hydrostatic bearings in which oil is fed at high pressures into the gap from an independent source. In these bearings the rubbing surfaces are separated by an oil film even before the machine is actually started; the variation of the machine speed in no way affects the lubrication of the bearing or, in other words, its serviceability.

An example of a hydrostatic support (step bearing) is shown schematically in Fig. 7. Oil from a pump flows through choke *1* into pocket *2* with confining edge *3*. The pressure in the pocket depends on the ratio of the cross-sectional area of the choke to the changing areas of the cross section between the confining edge and the verti-

cal journal of the bearing. As the load on the beairng increases this cross section becomes smaller and the pressure in the pocket rises, reaching at its maximum the pressure built up by the pump. In the case of impact loads the pressure in the pocket may by far exceed the pump pressure because of the blocking up of the choke as a result of a sharp increase in its hydraulic resistance.

In full-journal bearings loaded by alternating forces use is made of a system of radially arranged pockets (Fig. 8). In Fig. 8 the lower pocket is the supporting one. There is no pressure in the upper pocket because of an enlarged clearance along the upper arc of the bearing. The pressures in the lateral pockets are mutually equalized and so these pockets do not take up any load. The oil flowing out of the upper and lateral pockets functions as a conventional coolant.

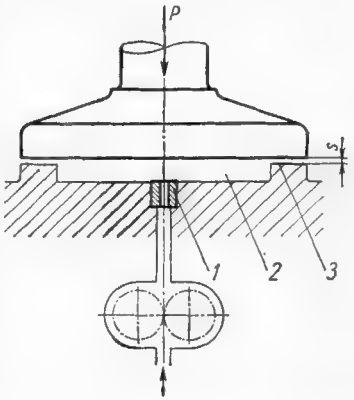


Fig. 7. Hydrostatic step bearing

Apart from the hydrostatic taking up of the load, a certain hydrodynamic effect also occurs. The oil flowing through the upper and lateral pockets is entrained by the rotating shaft and forced into

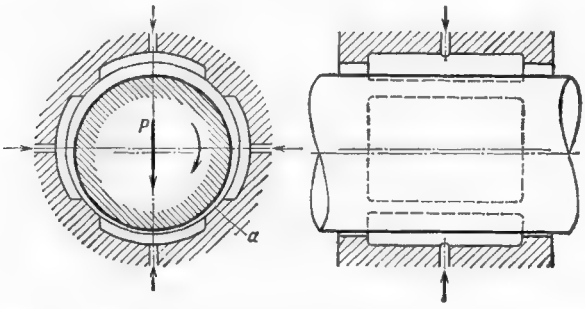


Fig. 8. Four-chamber cylindrical hydrostatic bearing

the converging wedge-shaped gap along the lower arc *a* of the bearing, building up an increased pressure on the surface of the confining edges and also in the supporting pocket (because of the hydraulic blocking up of the choke).

With the load direction altered through 180°, the upper pocket becomes the supporting one and the lower pocket, the feeding one. Similar effects occur when changing the load direction through 90°.

Thus, responding to the shaft movements, the bearing automatically adjusts itself to the taking up of the load in the direction of the acting force vector.

Recently for taking up alternating loads use has been made of honeycombed and porous hydrostatic bearings. These bearings operate similarly to the one above; in this case the honeycomb or pores act as pockets.

In certain cases (e.g., when dealing with high-speed spindles or guideways of metal-cutting machine tools) it is advantageous to employ aerostatic or gas-static lubrication, in which case the rubbing surfaces move on an air (gas) cushion built up by the air (gas) forced into the clearance between the surfaces.

In view of the development of hydrostatic bearings there occurs a revaluation of the comparative merits of sliding and rolling bearings, a certain preference being up to now given to the bearings of the latter type. The sliding bearings, when provided with proper lubrication, are in principle more advantageous since they totally exclude metal-to-metal contact and assure wear-free operation, whereas in rolling bearings such contact and wear are inevitable.

The employment of hydrostatic bearings, however, is limited because of their more elaborate lubrication systems and, in particular, because of the necessity for separately driven oil pumps during start and stop periods.

One of the most frequent causes of premature machine failures is corrosion. To preclude this, highly effective means of protection must be provided, especially for machines working in the open, under conditions of high humidity, or in chemically active media. Such means include electroplating (chrome, nickel or copper plating), deposition of chemical films (phosphating, oxidation), deposition of polymer films (capronization, polythenization).

Yet, the most advantageous solution is the use of corrosion-resistant structural materials (stainless steels, titanium alloys). Slightly loaded machine components operating in contact with chemically active agents should be made of chemically stable plastics (polyolefins, polyfluoroethylene resins).

The application of the above described technological and constructional means may allow the service life of the majority of components in general-purpose machines to be increased to practically any value required to ensure the necessary durability of the machines as a whole.

When working on a new project, designers often do not plan the durability of components but only select their shapes, sizes and machining methods, following the traditions and standards existing in a certain branch of engineering, which in new conditions of continuously intensified operation of machines and in the light of new ideas about the import of durability must be revised. In most

cases it is sufficient to clearly outline the task and apply general rational design processes so as to solve already at this stage many a durability problem that otherwise would have to be solved by way of refining the design of the already built machine, which is waste of time and effort.

Like in the aircraft industry where each component at its design stage is subjected to a careful check for weight, in the general machine building it would undoubtfully be a good practice to subject all parts to a systematic durability control.

Naturally, there are exceptions from the general rule. It is very difficult to impart durability to parts running in direct contact with abrasive media (impellers of pumps handling slurries, soil-cultivating implements, cutters in mining machines, caterpillar tracks, jaws of stone-crushers, chains and drives of countinuous conveyers for handling cement, coal, etc.).

The service life of such components in some instances (e.g., rock bits) is only tens of hours and can be prolonged only by selecting the most wear-resistant materials and applying the most effective strengthening processes.

Of course, measures taken for improving the durability make the design more expensive. They will mean the use of high-grade materials, introduction of new technological processes and, occasionally, establishment of new sections in a factory. The additional capital investments not uncommonly daunt managers who consider costs in the enterprise and not the effect of the machine's durability and reliability on the national economy. Such expenditures are justified. The cost of producing components determining the durability of a machine is negligible in comparison with the machine cost and the latter, generally, is rather small when compared to the total sum of operational expenses. Inappreciable in the total cost balance, the above-said additional expenditures on improving the durability will eventually bring huge financial gains because the downtimes and cost of repairs are reduced.

Hence an important practical conclusion: trying, as a general rule, to make the machine cheaper, one should not economize on the manufacture of components determining the durability and dependability of the machine. One should never spare expenses on researches associated with the development of new, better materials and processing methods that would enable a greater durability to be obtained.

Many manuals on mechanical engineering recommend the use of the cheapest materials and simplest manufacturing methods fit for the function of a given part. This recommendation cannot be accepted without reserve. The selection of materials and production techniques must always be based on the assessment of the relative

weight of the additional manufacturing costs in the totality of operational expenditures.

All components on which the durability and reliability of machines depend should be made of the highest-grade materials and by the most perfect processing methods.

Let us consider, as an example, the manufacture of piston rings used in internal combustion engines and in other piston-type machines. The quality of these rings largely predetermines the engine maintenance cycle. As the rings wear down the engine efficiency falls and the fuel and oil consumption increases. At present the service life of such rings amounts to 500-1000 hours. The use of the latest technological procedures which increase the wear resistance of the ring-cylinder pair (porous chrome plating and sulphocyaniding of the rings and nitriding of the cylinder faces, etc.) makes it possible to prolong the service life up to 5000-10 000 hours. The insignificant rise of the engine cost, due to the more costly rings, is repaid many times because of the increased wear resistance of the engines which are now so common (in automobiles, tractors, diesel locomotives, ships). As a result, a great economic effect is provided for the national economy.

Another example are antifriction bearings. Most authors recommend the use of less accurate bearings, arguing that the higher the bearing accuracy, the greater their cost.

Let us assume that the manufacturing cost of the class H bearings (ordinary accuracy) is one unit. Then, the manufacturing costs of bearings can be expressed by the following proportionate figures (in the order of rising accuracy): improved (class П)—1.3; extra-improved (class ВП)—1.7); high (class В)—2; extra-high (class АВ)—3; precision (class А)—4; extra-precision (class СА)—7; and super-precision (class С)—10.

At first sight these figures rather convincingly speak in favour of the use of the less accurate bearings.

However, such a conclusion is short-sighted. As the wear and damage of antifriction bearings is one of the commonest causes of machine failures, which to a significant degree predetermines the in-between repair periods, one must acknowledge that it is more sensible and economically profitable to employ bearings of higher accuracy in spite of their higher cost. Of course, this does not mean that precision bearings must always be used, and it does not liberate the designer from his responsibility for ensuring the durability of the bearings by their correct installation and lubrication.

Limits to increasing the durability. The effectiveness of increasing the durability, as a means of enlarging the actual size of the machine fleet, lowers with the increase of the absolute durability values. With a successive rise of the durability, each year added to the absolute durability value gives an ever-decreasing gain in

the size of the operating machine fleet as compared to the previous year.

The graph in Fig. 9 shows the change in the relative size of the machine fleet with the rise of the durability of the model being manufactured. The durability of the original model is assumed to be equal to one year. Increasing the durability by one year doubles

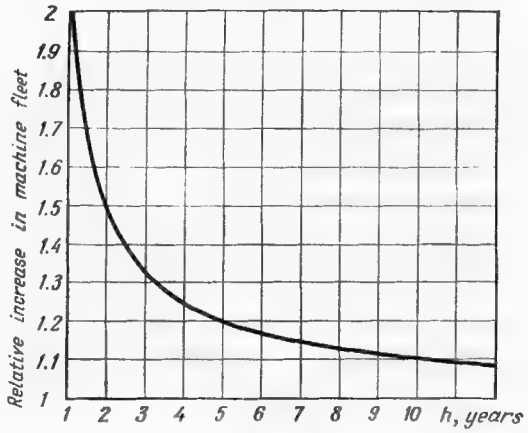


Fig. 9. Relative growth of machine fleet with increase of machine durability (size of machine fleet with initial durability $h = 1$ year is taken as unity)

the size of the machine fleet. With the durability increased by as much again, the gain relative to the previous model comes to 1.5 times (though in reference to the original model it is 3 times). Increasing the durability by another year brings the size of the machine fleet up to 1.33 times that for the previous model (though in reference to the original model the gain is 4 times). With each yearly increase in the durability the size of the machine fleet grows less and less. Hence, it is important to set the most reasonable limit to the durability increase, which will give a significant gain in the size of the machine fleet without causing an excessive rise of the model cost. In the case illustrated by Fig. 9 the growth of the machine fleet practically stops when the durability increase reaches 5-6 times.

The magnitude of technically obtainable durability largely depends on the loading rate of the machine.

For transport machines the durability totals 10 000 to 20 000 hours, and the service life, 5 to 8 years. For stationary machines, e.g., machine tools, the durability comes to 50 000-100 000 hours, which corresponds to 15-25 years of service to a double-shift schedule and 10-20 years with a triple-shift operation. At such service life periods the problem of obsolescence becomes very urgent.

The mechanical life of machines can be artificially prolonged by way of restoration.

However, from the standpoint of economy this way is inadvisable, as the cost of repairs often exceeds by many times that of the original machine.

At the initial stage of operation the repair expenditures are usually small but later on they sharply grow as the need for routine and medium repairs arises. Eventually, when the machine is subjected to a capital repair the total repair costs reach a considerable value comparable to the machine cost. The question of whether a machine is worth further use should be settled just before submitting the machine to an overhaul. Putting aside for the present the problem of obsolescence, it is economically advisable to consider the limit to the machine use to be the moment when the impending capital repair expenditures approximate the cost of a new machine. Then it is unquestionably more advantageous to buy a new machine than to repair the old one. The more so as new machines are of higher quality than repaired ones and have better characteristics thanks to continual technical improvements. Moreover, there is a regular lowering of the cost of new machines due to the undeviating intensification and improvement of industrial processes.

The total sum spent on all repairs should also be considered when deciding the question of the termination of a machine use: as a general rule it may be said that the total sum spent on repairs during the machine life should not exceed the machine cost.

Attempts have been made to find optimum durability, i.e., the durability with which the cost of the products turned out by the machine is minimal. One proceeds from the following prerequisites: the cost of production is equal to the sum of fixed expenditures independent of the duration of the machine use (expenditures on power, materials, labour, etc.) and variable expenditures dependent on the duration of the machine use (depreciation expenditures which are inversely proportional to the machine use duration, and repair expenditures which grow larger with time because of the wear of the machine).

The variation of production costs Cp as a function of the duration of the machine service is expressed by the following formula

$$Cp = Ef + \frac{C}{H} + Rpr = \varphi(H) \qquad (1.20)$$

where Ef = fixed expenditures;
C = machine cost;
H = duration of service;
Rpr = cost of repairs

Summing up the above constituents gives the costs of production as a function of D (thick line in Fig. 10*a*). The cost curve has its

minimum, and it is suggested that the durability corresponding to this minimum be considered optimum.

This interpretation is too simple to be used in practice. Firstly, in most cases the variable expenditures are negligible as compared to the fixed ones, so that even when the cost curve has a minimum,

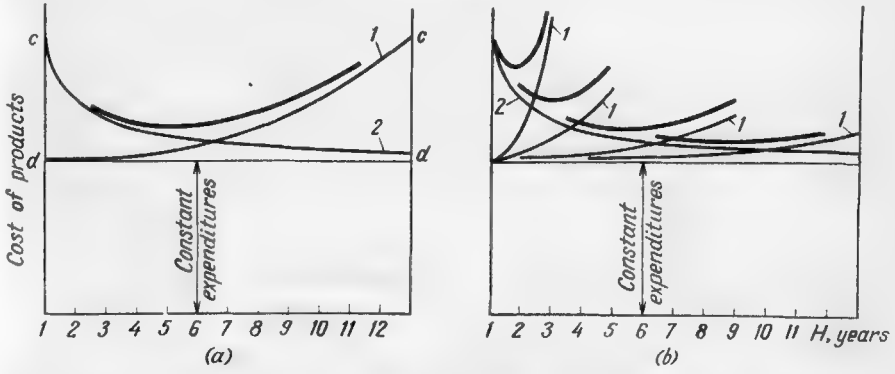


Fig. 10. Machine cost (heavy lines) as a function of machine's service life H (*a*) with $Rpr = 4C$; (*b*) with $Rpr = C$; *1* — annual cost of repairs; *2* — depreciation expenditures

the latter is too weakly expressed. The machines which have exceeded the optimum durability term are still capable of turning out products for a long time, though at a somewhat less profitable rate.

Secondly, the total cost of repairs $\sum Rpr$ throughout the entire period of the machine service is not accounted for. Thus, for the case depicted in Fig. 10*a*, where the repair expenditures for the last year of service are assumed to be equal to the cost of the machine (lines *cd*), the total repair expenditures (the area between the repair cost curve and fixed expenditures line) equals four times the machine cost, which is certainly an overestimated figure.

The picture becomes different after setting sensible limits on the total cost of repairs. Thus, if one takes these costs not to exceed the machine cost, then there will be a definite minimum product cost (thick curves in Fig. 10*b*) for any given length of the machine service. These minimums decline and become less clear as the service life increases. As the service life grows longer, the envelope of these minimums continuously falls down. Thus, with the total repair costs confined within certain limits, the concept of optimum durability vanishes; the costs of production continuously decrease.

Finally, it should be mentioned that the discussion on the concept of optimum durability neglects the dynamics of changes in the fixed expenditures which, generally, tend to reduction (decrease of the

costs of power, materials and labour thanks to automation and improvement of processes). This reduction may still more change the picture in favour of longer service lives.

(*f*) *Durability and Obsolescence*

The problems of increased durability and obsolescence are intimately associated. A machine becomes obsolete when its performance characteristics are no longer fit to satisfy the production needs because of higher requirements, or of more perfect machines having made their appearance, although the machine itself still retains its mechanical capabilities.

The signs of obsolescence are as follows: degradation (compared to the mean value) of reliability, product quality, and productivity; increase of power consumption per unit product, costs of labour, maintenance and repairs, and, as a general result, reduction of the machine's profitability.

Obsolescence is not necessarily connected with physical wear (although, physical ageing impairs the machine performance, thereby expediting obsolescence). A machine may become obsolete while being quite sound mechanically and even new.

The main effect of obsolescence is reduced productivity per unit of labour force, which is the main indicator of economic progress.

Undoubtedly obsolescence will ensue in two cases: when changing over to a new product (a complete change of the technological process), and with the development of new processes or new designs allowing machines better than those of the old type to be made.

The latter kind of obsolescence is exemplified by the development of turbojet engines which in aircraft ousted almost fully the older internal combustion engines (piston engines).

It must be noted, however, that such radical and prompt changes are rather uncommon. Under the conditions of gradually improving technology the problem of obsolescence is otherwise.

To begin with, in most cases, particularly in heavy-duty machines, physical wear by far outpaces obsolescence. For example, the physical reserves of automobile trucks operated under heavy conditions are exhausted in 5-6 years, although their technical and economic ratings provide for a much longer service life.

Secondly, efficient methods of preventing machine obsolescence exist.

The most important of these methods is a prospective machine design, i.e., the design considering the dynamics of changes in that particular branch of industry for which the machine is intended. The design of the original model must be capable of further development in respect to its production capacities, power, output and versatility. Such a provision will in due time allow the necessary modi-

fications to be made, thus keeping the machine ratings on a par with the growing demands without retirement of the basic model and, consequently, avoiding the construction of new production lines which is inevitable when switching over to a new model.

With machines being already in use, the presence of reserves makes it possible to intensify the operation of the machines as the demands of production grow higher.

Another method of preventing obsolescence is to use the machine more fully in actual service. The shorter the period during which it will exhaust its durability reserves, i.e., the closer its service life to the mechanical life, the higher is the guarantee that the machine will not become obsolete. The reduction of the service life of machines to 3-4 years will practically guarantee them from the risk of obsolescence.

The reduction of the machine's service life by no means implies a decline in the production output. As shown earlier, the total output of a machine depends not on its service life, but on its durability.

The reduction of the service life with the durability remaining constant is essentially effected by intensifying the use of the machine.

With regard to machines operated to a calendar-based schedule the problem is solved by increasing the number of working shifts and enhancing the loading level.

In design this problem is solved by making the machine as universal as possible, i.e., by extending its range of operation, and, primarily, by enhancing its reliability, which results in shorter emergency and repair downtimes.

The degree of utilization of machines of aperiodic action (e.g., seasonal machines) can be raised by applying changeable trailed or mounted equipment, which increases their working period in a year.

The rate and degree of obsolescence depend on the scale and technical level of production. At large industrial enterprises which are capable of accelerating the production and continuously improving the working processes machines become obsolete earlier, in contrast to those employed at medium and small enterprises developing more slowly.

Machines which are considered obsolete in conditions of advanced production can be used to advantage either at less important sections of the same plant or by smaller enterprises having insufficient equipment where these machines will successfully contribute to the output. Although such machines will somewhat lower the rate of the total productivity rise they will keep delivering products until their mechanical life is fully exhausted.

Proceeding from the above one may say that obsolescence is not the absolute limit to the durability increase. This limit can either be moved farther away by rationally selecting the design parameters of the machine or entirely eliminated by intensifying its use. Conse-

quently, obsolescence cannot be regarded as a serious argument against enhancing durability. Naturally, this by no means implies that the designer may neglect the obsolescence factor; on the contrary, he must do everything to avoid it.

1.4. Operational Reliability

The following are the reliability features of a machine: high durability, trouble-free operation, stability of action (ability to maintain original parameters over long periods), endurance (ability to sustain overloads), simplicity of maintenance, survivability (capability of operating for some time after partial failures, even if with poorer characteristics), repairability, long periods between repairs and, finally, small scope of repair work.

Because of the many factors influencing the reliability it is rather difficult to formulate a single reliability criterion. When defining reliability one most often proceeds from the concept of machine failure, i.e., any forced stoppage of the machine.

The reliability of a machine may be characterized by:

frequency of failure;

duration of an uninterrupted operation of the machine between failures;

regularity of changes in the frequency of failure during the machine's being in operation;

severity of failures and the scope, cost, and duration of repair work necessary to eliminate them.

The duration of the forced stoppages of a machine is characterized by the downtime factor η_{dt} (in other words, faultiness factor) which is the ratio of the total duration of downtimes h_{dt} over a certain period to the sum of the actual working time h_{act} and downtimes h_{dt} over the same period

$$\eta_{dt} = \frac{h_{dt}}{h_{act} + h_{dt}} = \frac{1}{1 + \frac{h_{act}}{h_{dt}}}$$

The duration of a trouble-free operation of a machine is characterized by the faultlessness factor

$$\eta_{fault} = \frac{h_{act}}{h_{act} + h_{dt}} = 1 - \eta_{dt}$$

According to their severity all failures are subdivided into light, medium and heavy.

Light failures are those which can be eliminated on the spot by the operating personnel and by means of the available hand tools.

Medium failures require that the machine be stopped for a period of time sufficient for a partial disassembly and change (or recondition) of faulty parts, this being accomplished by repair services.

Heavy failures are such breakages as affect vital machine parts, necessitating a prolonged stoppage of the machine for the purpose of repairs. General wear of the machine can also be regarded as a heavy failure for at a certain stage it requires the complete overhaul of the machine and replacement of badly worn parts.

As to the origin of failures, there are distinguished failures due to design and manufacturing defects, failures due to maloperation, and incidental failures.

Maloperation implies negligent attendance, violation of operating instructions, disregard of prescribed operating conditions (overloads), mistakes in the order of executing control operations (wrong engagements), noncompliance with safety regulations, etc.

Most failures attributed to maloperation may be, with good reason, put down to design defects. A well-designed machine must incorporate features which preclude its operation with dangerous overloads, prevent conflicting engagements and reduce to a minimum the effect of the quality of maintenance on the serviceability of the machine.

It would be possible to characterize the reliability of a machine by the scope of work necessary to eliminate failures, i.e., in the final analysis, by a repair cost index which characterizes in an aggregatory manner the frequency and degree of failures and the repairability of machines. But this index is relative for the followingreasons. Firstly, the in-between repair intervals and cost of repairs depend not only on the machine's reliability, but also on the quality of maintenance and repair. Secondly, the cost of repairs is influenced not only by the scope of the repair work required, but also by the organizational level of the repair services. Thirdly, the total cost of repairs depends on the policy of increasing the machine's service life. When following a wrong policy of restoring machines instead of increasing the manufacture of new machines and, particularly, their durability, the repair expenditures may reach formidable values.

The theory of reliability is at present being developed. Its objectives are as follows: determination of requirements for reliability from the standpoints of technology and economy; study of statistical laws governing failures; investigation into causes of failure (failure diagnosis); detection of parts and units most frequently causing failures; prediction of failures; assessment of the severity of failures and the difficulty of their elimination; study of failure effects upon the economical aspects of machine operation; development of reliable indices characterizing the reliability of machines.

Special sections of the theory deal with the joint reliability of complex sets of machines (flow and continuous production, semi- and fully-automatic lines, transfer machines, etc.). These sections treat the following problems: control of such complex sets of machines and monitoring of their interaction; redundancy (introduction of stand-by machines and circuits); accumulation (introduction of magazines

which assure an uninterrupted operation of a group of machines in the event of a short-time failure of some machine in the group), etc.

The theory of reliability in many respects is closely associated with the theory of durability, as far as their essence and methods are concerned. Like the theory of durability, the theory of reliability is based upon the methods of the theory of chances and mathematical statistics and its conclusions are formulated as probability relations.

The reliability of a machine is characterized by the mean-probable time of an uninterrupted operation (mean-probable output of the machine per failure) as a function of the duration of service, or by the mean-probable frequency of failure, and also by the density of distribution of failures throughout the service life of the machine.

The theory of reliability which predicts failures most frequently occurring in practice may serve as a valuable aid to a mechanical engineer. The main active role of the latter is the elimination of weak parts in the design, thus raising the reliability of the machine as a whole. By applying all modern design methods and technological procedures, the designer can, in principle, (at least in many categories of machines) eliminate all failures, except those occurring by pure chance.

(*a*) *Means for Improving Reliability*

The reliability of a machine in the first place is dependent on the strength and rigidity of its structure.

The following methods improve the strength of the machine without increasing its weight: application of the most advantageous sections and forms, maximum utilization of the strength of materials and uniform, as far as possible, loading of all the elements of the structure.

Rigidity can be increased by correctly selecting the loading patterns, rationally arranging the supports, and giving a stiff shape to the structure.

The faultlessness of operation and the duration of in-between repair periods depend to a large extent on the correct use and careful maintenance of the machine, timely preventive repairs and prevention of overloads. Yet, it would be wrong to wholly depend on the quality of maintenance. The prerequisites for correct operation must be built into the machine design. Reliable operation must be assured even in conditions of unsatisfactory maintenance. If the machine fails because of unskilled hands it means that the construction was not designed correctly from the standpoint of reliability.

The human factor in the maintenance and operation of a machine should be eliminated as far as possible and the maintenance operations should be minimized.

Subject to elimination are periodic operations, such as readjustments, tightening-up operations, lubrication, etc., which in the case of unskilled servicing may lead to an excessive wear and premature failure of the machine.

For instance, in internal combustion engines the necessity for the adjustment of clearances in the valve mechanism can be eliminated by means of automatic devices compensating for wear and thermal expansion (hydraulic or some other devices). Apart from simplifying the maintenance, such compensating devices practically eliminate all the clearances in the valve mechanism, thus materially enhancing its durability.

Periodic adjustments of the main and crankpin bearings can also be obviated. The modern level of lubrication techniques allows such bearings to be designed as will operate practically infinitely and with a minimal wear.

The necessity for a periodic tightening up of nuts and bolts, which work loose in service, can be dispensed with by using modern self-locking screw connections.

Machine operation can be heavily complicated by an inadequate lubrication system requiring much attention of the maintenance personnel. Intermittent point lubrication should be avoided. When, because of design features, this is not feasible, it is advisable to use self-lubricating bearings or introduce a centralized system which feeds oil to all the lubrication points from one station.

As far as dependability and convenience of operation are concerned, the best solution is to introduce a fully automated lubrication system which does not require periodic oil changes. This will be feasible, if measures are taken to prevent the oxidation and thermal degeneration of oil and if means are provided for continuous oil refining and regeneration.

Lubrication systems must incorporate emergency devices which ensure at least a minimum delivery of oil in the event of a failure of the main system.

One of the ways to enhance operational reliability is to duplicate servicing devices which most frequently fail in the course of work. This can be exemplified by a dual ignition system of petrol engines or a dual automatic control system. Multiple duplication of control systems is exercised where a failure may endanger human lives. Of great significance is automatic protection against accidental or preconceived overloads. Such a protection is effected by safety and limiting devices operating in a tracking mode and coming into action as soon as an overload occurs.

Fully automated control is most expedient since it turns the machine into a self-servicing and self-regulating unit which adjusts itself to optimum operating conditions.

This can be illustrated by self-changing gearboxes and automatic transmission systems of automobiles which provide for a stepless control of the transmission ratio from the engine to the road wheels. Such a system adjusts automatically to the optimum transmission ratio which suits the given road and running conditions, thus increasing the engine efficiency and life.

High reliability can only be achieved by implementing a whole complex of design, technological, and organizational measures. The increasing of reliability requires a prolonged, daily, scrupulous, purposeful, combined work by designers, process engineers, metallurgists, researchers and practical engineers, done to a carefully developed programme.

The prerequisites for the manufacture of high-quality products are: progressive manufacturing procedures, high level of production, rigorous adherence to the prescribed processing conditions, and careful control of products at all stages of manufacture, beginning with the manufacture of component parts and ending with the assembly of the finished products.

Objective evaluation of the indices of reliability, durability and operational costs presents the task of great difficulty. These indices can only be determined for certain upon expiration of a long period of time, with the products being sent far away from the parent factory and dispersed among various remote operational points.

Under such circumstances the accelerated test methods for determining the durability of parts, units, assemblies and machines as a whole assume great importance. Great assistance in this respect can be rendered by durability research laboratories carrying out systematic tests on the wear and service life of finished products.

A wider use should be made of the method of simulating actual working conditions, which consists in the bench and operational testing of machines under conditions much more severe than those of the normal service. Thus, the machine in a very short period of time undergoes a test which under normal conditions would take several years. Tests are continued until the wear limit of the machine is reached or even until its full or partial breakage occurs. At times they are stopped for the degree of wear to be measured, the state of parts registered, and tokens of impending failure determined.

Similar vigorous tests enable structural defects to be found and the necessary precautions for their exclusion taken. Accelerated tests also provide sufficiently reliable starting information for assessing the actual machine's durability.

Refinement of machines in operation. To develop reliable and durable machines it is necessary to carefully study the experience gained in their operation. The work of designers on a machine must not terminate in the State tests of the prototype or in putting the machine into serial production.

The final refinement of the machine begins, in fact, after it has been commissioned. Testing the machine in actual service is the best way to find and then eliminate the weak spots in its design.

The disadvantages of a machine are especially clearly revealed in repairs. Therefore close and continuous connections must exist between the designers and maintenance departments. For factories engaged in mass or multiple production it is convenient to have their own repair sections functioning as study laboratories and schools for better design.

Of interest is the system of repair organization which is in practice at engineering plants in the United States. According to the data obtained during a survey of 356 American firms by the Institute of Economy of the USSR Academy of Sciences, in 46% of cases the chiefs of design departments are simultaneously in charge of repair shops; in 18% of cases the repair shops are subordinated to the chiefs of design departments; in 24% of cases—to other services, and only in 12% of cases are repair shops independent sections. Thus, in 64% of cases repair shops are, in one form or another, directly linked with design departments. This tendency is not accidental but the result of a definite policy of linking the design and repair services closer together so that the quality of products can be enhanced.

When examining defects it is absolutely necessary to discriminate between random and systematic defects. Random defects usually result from unsatisfactory control or technological inadequacies at the manufacturing plant, while systematic defects are an evidence of poor design requiring immediate rectification.

Operational observation of a machine should be included into the work plan of design offices along with the design work and occupy a significant part of the designer's time. When isolated from the actual operation of the machine, the designer cannot advance and will never come to the top in his profession.

1.5. Machine Cost

Lowering the cost of mechanical-engineering products is a complex task, both design and manufacturing. Sharp decreases in the cost of manufacturing the machines can be obtained through rationalization of production (mechanization and automation of production processes, concentration of processing operations, specialization of plants, industrial cooperation, etc.).

These measures are practicable and have the greatest effect in the case of a stable large scale production. First and foremost here is the designer. He must incorporate in the design prerequisites for the production of one model over a long period of time and on the largest scale possible, i.e., develop a design possessing wide applicability and capable of future development and modernization.

Of great importance is the reduction of the number of the machine type-sizes through a rational selection of the machine types and parameters, as this makes it possible to manufacture the machines on a multiple production basis and thus cut down their cost.

The suitability of the design for industrial production is no less important. Under the suitability for industrial production we understand the combination of features which ensure the most economical, prompt and efficient manufacture of machines by advanced techniques, parallel with the higher quality, accuracy and interchangeability of parts.

This concept should also cover the features providing for the most efficient assembly (suitability for assembly) and the most convenient and economical repairs (suitability for repairs).

The suitability for industrial production depends on the scale and kind of the production. The piece and small-lot production requirements as to the suitability of the design differ greatly from the multiple and mass production requirements. The features characterizing the suitability of the design for industrial production are not the same for different machine components.

Large economic gains are obtained from the unification and standardization of parts, units and assemblies.

(*a*) *Unification*

Unification consists in a repeated use of the same elements in the design, which enables the range of parts to be reduced, cuts down the cost of manufacture and facilitates the operation and repairs of machines.

The unification of constructional elements makes it possible to curtail the number of processing, measuring and mounting tools. Subject to unification are connection joints (by diameters, classes of fits and accuracy), threaded connections (by diameters, types of threads, classes of fits and accuracy, sizes of jaw), key and spline connections (by diameters, shapes of keys and splines, classes of fits and accuracy), gears (by modules, teeth types, classes of accuracy), chamfers and fillets (by sizes and types), etc.

Unification of originally designed parts and units can be internal (for a given product only) or external (adoption of components of other machines produced at the same or adjacent factories).

The greatest economic gain is obtained when components already being produced on a mass scale are used in new machines. The adoption of components of individually produced machines, or of old machines already discarded or about to be discarded, and also of machines produced by other enterprises wherefrom the procurement of the components is difficult or even impossible has only one

advantage: the successful use of such parts has been practically verified. Not uncommonly this is enough to justify unification.

The unification of grades and assortment of materials, electrodes, sizes and types of fasteners and other standardized parts, antifriction bearings, etc., eases the supply of materials, standards and purchased parts to the manufacturing plant and repair departments.

The degree of unification is estimated by factor η_{unif} which is the ratio of the number of the unified components to the total number of components in a given product

$$\eta_{unif} = \frac{z_{unif}}{z} \cdot 100\%$$

or the ratio of the weight of the unified components to the total weight of the product

$$\eta_{unif} = \frac{G_{unif}}{G} \cdot 100\%$$

or else the ratio of the cost of the unified components to the total cost of the product

$$\eta_{unif} = \frac{C_{unif}}{C} \cdot 100\%$$

The disadvantage of the first index is in that it does not take account of the specific value of the unified components in the machine construction. The second index allows only for the weight of the unified components in the total weight of the machine. The third index is most correct, but it is more difficult to determine than the former ones.

The degree of internal unification can be evaluated by the coefficient of recurrence

$$\eta_{rec} = \left(1 - \frac{N_n}{N_p}\right) \cdot 100\%$$

where N_n = number of differing names of parts in the product;
N_p = total number of parts in the product

This coefficient is readily determined from the summary parts list and characterizes the degree of perfection of the design in respect to the reduction of the range of parts. In well-designed machines $\eta_{rec} =$ = 40-60%.

For a differential assessment the following indices are applied.

The degree of unification of original components

$$\eta_{unif.orig} = \frac{N_{unif.orig}}{N_{orig}} \cdot 100\%$$

where $N_{unif.\ orig}$ = number of the originally designed components unified;
N_{orig} = total number of the originally designed components

The degree of unification of constructional elements

$$\eta_{elem} = \left(1 - \frac{N_{t\text{-}s}}{N_{elem}}\right) \cdot 100\%$$

where $N_{t\text{-}s}$ = number of type-sizes of given elements;
N_{elem} = total number of given elements in the product

For example:

the degree of unification of threads

$$\eta_{thrd} = \left(1 - \frac{N_{thrd.t\text{-}s}}{N_{thrd}}\right) \cdot 100\%$$

where $N_{thrd.\ t\text{-}s}$ = number of thread type-sizes;
N_{thrd} = total number of threaded connections in the product

the degree of unification of fasteners

$$\eta_{fst} = \left(1 - \frac{N_{fst.t\text{-}s}}{N_{fst}}\right) \cdot 100\%$$

where $N_{fst.\ t\text{-}s}$ = number of fastener type-sizes;
N_{fst} = total number of fasteners in the product

(*b*) *Standardization*

Standardization is the regulation of the designs and type-sizes of components (fasteners, bushes, pipe fittings, unions, nipples, oilers, seals, control elements, such as handles, levers, knobs, wheels, etc.) and units and devices (couplings, cocks, shutters and gates, lubricating devices, pumps, filters, pressure reducing valves, air- and fluid-actuated units, etc.) which are widely used in mechanical engineering.

All existing industrial standards are classified as National, branch (pertaining to individual industries) and departmental. Nearly every specialized design organization standardizes parts and units which are typical of a given branch of mechanical engineering.

Standardization hastens the design work, facilitates the production, operation and repairs of machines. A correct choice of the design of the standardized parts increases the machine's reliability.

Standardization produces the greatest effect when the number of normally applicable standard sizes and types is reduced, in other words, in the case of their unification. The design organizations practically solve this problem by issuing the so-called limiting lists covering the minimum range of standard items, which satisfies the needs of the given class of machines.

The advantages of standardization are fully realized when the standard parts are manufactured in a centralized fashion at specialized factories. This relieves machine-building plants of the laborious

work of manufacturing standard parts and simplifies the supply of spares to repair enterprises.

The degree of standardization is evaluated by the factor

$$\eta_{stand} = \frac{N_{stand}}{N_t} \cdot 100\%$$

where N_{stand} = number of standard parts;
N_t = total number of parts in the product

For the successful accomplishment of standardization the standard parts must be of high quality. Furthermore, the application of standard parts should in no way restrict the creative initiative of the designer and prevent him from finding better, more rational solutions. The designer should not refrain from the application of new solutions in the fields already covered by standardization, provided that these solutions have obvious advantages over the existing standards.

1.6. Building up Machines Derivatives on the Basis of Unification

Unification is an effective and economical means for building up on the basis of an original model a number of machines of the same purpose, but having different characteristics as to their power capacity, productivity, etc., or machines of different purposes, performing qualitatively different operations, and also those intended to turn out other products.

At present this problem is solved in several ways not all of which are universal. In most cases each method is applicable only to certain machines categories, the economic effects of these methods being also different.

The classification of the methods for building up unified machines derivatives, given below, is arbitrary. Some of the listed methods are so close together that it is rather difficult to draw a clear demarcation line between them.

A combined and parallel use of two or more methods are also possible.

(*a*) *Sectionalization*

This method consists in dividing a machine into identical sections and building up machines derivatives by stacking the required number of the unified sections.

Sectionalization is applicable to many types of materials handling machinery (belt, scraper and chain conveyers). Sectionalization in this case comes to building up machine frameworks from unified sections and making machines of various lengths with a new carrying belt.

Particularly easy to sectionalize are machines with a link-type carrying belt (bucket elevators, platform conveyers with roller chains) in which the length of the belt can be varied by removing or adding links.

The economic efficiency of this method is little affected by the need for introducing some non-standard sections which may become necessary to make the machine length suit local conditions.

Suitable for sectionalization are also plate filters, plate heat-exchangers, and centrifugal, vortex and axial hydraulic pumps. In the latter case, by stacking different numbers of sections one can obtain a series of multi-stage pumps rated for different pressures, with their main working members being unified.

(b) *Method of Changing the Linear Dimensions*

This method enables the productive capacity of machines and units to be changed by varying their length without changing the cross-sectional shape. The method is applicable to a limited class of machines (mostly of the rotary type) whose capacity is proportional to the length of their rotors (gear, rotary and vane pumps, Ruth's compressors, agitators, etc.).

With this method the degree of unification is rather moderate because subject to unification are only the end plates of housings and some auxiliaries. The main economic gain is due to the fact that the basic equipment for machining the rotors and interiors of the housings remains unchanged.

Increasing the capacity of gear trains, speed reducers and gearboxes by extending the tooth length of the gears without changing their module is a particular case of the application of this method.

(c) *Method of Basic Unit*

This method enables machines of various purposes to be obtained from one basic unit to which special equipment is attached to suit a particular task. It is most popular in the construction of road-building machines, mobile cranes, loaders, pilers, snow-removing machines and also special-purpose self-propelled vehicles. In this case the basic unit is a mass-produced automobile or tractor chassis. Machines for different purposes are obtained by mounting auxiliary equipment on the chassis.

The basic unit method is extensively used in the manufacture of agricultural machinery.

The mounting of auxiliary equipment necessitates the development of additional devices and attachment mechanisms (power take-off boxes, hoisting and swivelling mechanisms, winches, reversing

and friction gears, brakes, control mechanisms, cabs, etc.). These additional mechanisms may, in their turn, also be unified considerably.

(*d*) *Conversion*

With the conversion method the basic machine or its principal elements are used to make different-purpose units, sometimes alike and sometimes different as to their working process.

As an example may serve the conversion of internal combustion engines from one kind of fuel to another, or from one type of the ignition process to another (from spark-ignition to compression-ignition).

Petrol engines are comparatively easy to convert into gas engines. This is achieved by replacing the carburettor with a mixer, changing the compression ratio (this is most readily done by changing the height of the pistons), and making some minor constructional alterations. The engine by and large remains the same.

Converting a petrol or gas engine to a diesel is a much more difficult job, chiefly because of the higher operating forces inherent in the diesel engines due to their high compression ratio and enormous ignition pressure. Consequently, the engine to be converted must possess significant safety factors. Conversion in this case consists in replacing the carburettor by an injection pump and injectors (or pump and injector units) and changing the compression ratio (replacing cylinder heads, increasing piston height or altering the configuration of piston heads).

Still another example of conversion is the change-over of piston air compressors to some other working fluid (gas, ammonia, Freon). Here account should be taken of the different physical and chemical properties of the new working fluids and select the materials for the working members accordingly.

A radical conversion is illustrated by the change of an internal combustion engine to a piston compressor. In this case the cylinder heads are replaced by valve boxes, which requires corresponding changes in the timing gear, thereby involving many alterations.

(*e*) *Compounding*

The method of compounding (parallel combination) consists in combining machines or units in parallel with a view to increasing the total power or capacity of the installation.

The machines to be combined can either be placed side by side, as self-contained units, or linked by synchronizing, transporting or other devices, or else designed as one integral unit.

An example of the combination of the first type is a twin-engine ship propulsion system with each engine driving its own screw, or

a twin- or multiple-mounted engines in aircraft wings. Apart from increasing the total power output (when it is difficult to build one engine of a sufficient power capacity), this method occasionally helps successively solve other problems. Thus, parallel-mounted engines in a ship add to its manoeuvrebility, particularly at low speeds.

Installation of several engines on aircraft contributes to better banking and steering. Several engines also enhance aircarft reliability: if an engine should fail the aircraft will still be capable of flying, although at a lower speed.

An example of the combination of the second type is a parallel (bank) installation of machine tools in groups, 2-3 pieces in each. Such installation is practised in transfer machines (lines), when the productive capacity of one machine in the flow line is considerably inferior to that of the whole line. In this case the component flow is divided into two or more streams (to suit the number of the machines installed in parallel) which are then again combined into a single flow.

An example of the combination of the third type is the doubling or trebling of the line-type machine tools, i.e., combining of several working routes on a single stand. As a result, there is a multi-line parallel-flow machine with its productivity increased in direct proportion to the number of the routes.

(*f*) *Modification*

Modification implies an alteration of a machine with a view to adapting it to other operating conditions, or other operations or kinds of products without changing its basic design*.

An example of modification is the adaptation of a machine for work in various climatic conditions. In this case required modification comes mainly to a change of materials. Corrosion-resistant materials are used in machines operating in humid tropical conditions while the machines intended for use in regions of severe cold are made of cold-resistant materials, with their lubrication systems being specially designed to assure trouble-free operation.

Another example is the modification of engines for use on marine transport. Here the task is to make the engine as lightweight as possible by replacing heavy alloys (cast iron) by light ones (aluminium alloys) and by introducing materials that will resist the corrosive action of sea water and air.

The modification of the machines intended for work in close contact with active chemical agents consists in protecting them against the adverse effects by introducing heavy-duty seals and using chemically-resistant materials.

* Occasionally the term modification may imply modernization of machines and improvement of their characteristics.

Modification is much more difficult when the machine has to be adapted to different operations or kinds of products. In this case the modification method comes very close to the multi-station machine system.

(g) *Multi-Station Machine System*

This method consists in building machines out of a varying number of standard self-contained units, or combinations of such units, which are mounted on a common base.

This principle is most fully embodied in the design of multihead metal-cutting machines. Such machines are built up of individual unified assemblies and elements (machining units, combination housings, synchronizing mechanisms, rotary tables, general-purpose housings, beds, pedestals, auxiliary units, coolant systems, electrical controls, hydraulic drives, etc.).

In the course of machining the workpiece generally remains stationary and the machining units are brought to it so that the machining takes place simultaneously on all sides, thus considerably speeding up the entire working process.

The chief advantages of the multi-station machine system consist in that it reduces the time and cost of the design and manufacture of machines, simplifies their servicing and repairs, and allows them to be easily reset for processing various components.

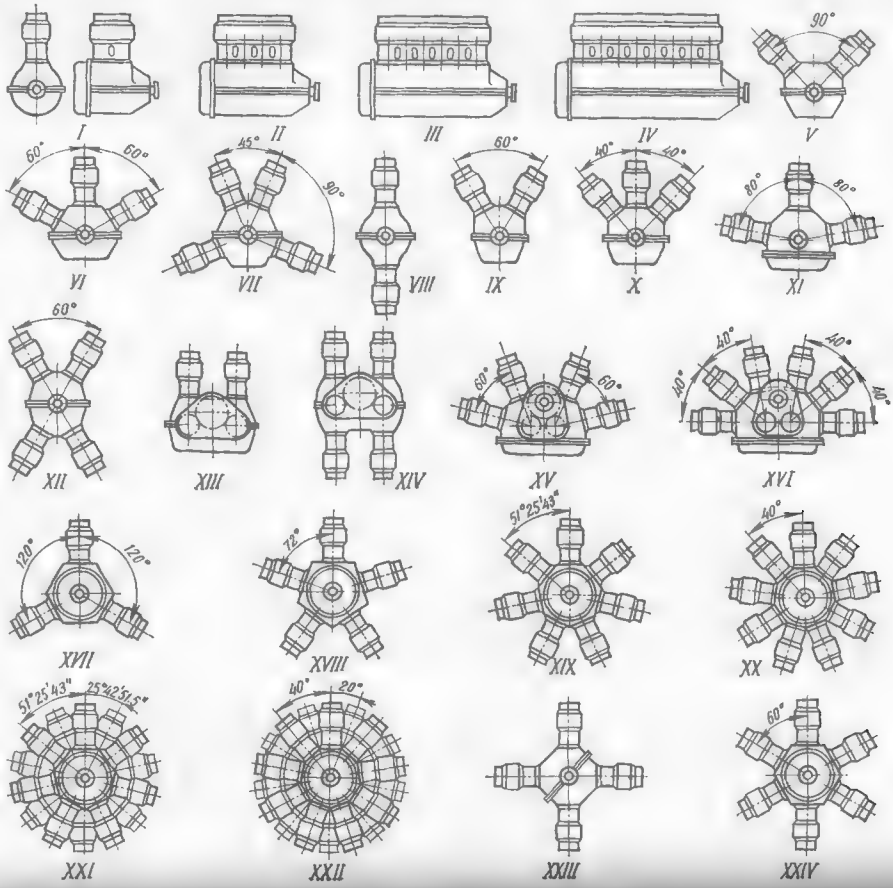
The method is rather promising and applicable to other machine tools.

A partial implementation of this method comprises the use of mass-produced standardized units (speed reducers, pumps, compressors), or the adoption of units and assemblies of serially-manufactured machines (gearboxes, differential gears, shifting mechanisms, clutches, friction couplings, etc.).

(h) *Integrated Standardization*

The method of integrated standardization is close to the previous one. It is successfully applied to simple units (vessels, settling tanks, evaporators, and mixture-preparing units which are extensively used in the chemical and food industries).

The structural simplicity of the above listed units enables all or nearly all of the elements of their design to be standardized. Suitable for standardization as to their sizes and types are the shells of tanks, bottoms, covers, manholes, hatches, fittings (valves, gates and shutters), fastening feet, stands. Subject to standardization are also whole assemblies (heat exchangers, agitator drives, metering devices, etc.).


I
II
III
IV
90°
V
60°
60°
VI
45°
90°
VII
VIII
60°
IX
40°
40°
X
80°
80°
XI
60°
XII
XIII
XIV
60°
60°
XV
40°
40°
40°
40°
XVI
120°
120°
XVII
72°
XVIII
51°25'43"
XIX
40°
XX
51°25'43"
25°42'51.5"
XXI
40°
20°
XXII
XXIII
60°
XXIV

Ref. No.	Engine type	Number of cylinders	Ref. No.	Engine type	Number of cylinders
I	Single-row	2	XIV	Four-row, twin crankshaft	24 (16)
II	Same	4	XV	Same	24 (16)
III	Same	6	XVI	Six-row, twin crankshaft	36 (24)
IV	Same	8	XVII	Single-row, radial cylinder	3
V	Double-row, V-type	8	XVIII	Same	5
VI	Broad-arrow, W-type	12	XIX	Same	7
VII	Four-row, X-type	16	XX	Same	9
VIII	Double-row, opposed cylinder	12 (8)	XXI	Double-row, radial, staggered cylinder	14
IX	Double-row, V-type	12	XXII	Same	18
X	Broad-arrow, W-type	18	XXIII	Row, cross-type	24 (16)
XI	Same	18	XXIV	Row, radial cylinder	36 (24)
XII	Four-row, X-type	24 (16)			
XIII	Double-row, twin crankshaft	12 (8)			

A specific feature of such apparatus is the wide use therein of purchased auxiliary equipment (pumps, vacuum pumps, filters, condensate traps, instruments and controls, automation means).

By using standard parts, unified assemblies, and purchased equipment one can build up:

apparatus with the same working process, but differing in size and productive capacity;

apparatus of the same purpose, but rated for different parameters of the working process (pressure, vacuum, temperature);

apparatus differing both in the purpose and working process.

(*i*) *Unified Series of Machines*

In some instances it is possible to form a series of machines derivatives of different power or productive capacities by changing the number of their main working members and using them in various combinations. Such a series is called a family, a range or just a series of machines. The method is applicable to machines whose power or productive capacity is directly proportional to the number of the working members.

The method provides the following technological and operational advantages:

simplification, acceleration and cheapening of the machine design and manufacturing processes;

possibility of applying highly efficient methods of machining unified parts;

shortening of the time required for testing and refining experimental models (because the main components have already been tested);

simplification of operation;

reduction of the training periods for specialists, shortening of repair times, and simplification of spares problem.

A classical example of the development of unified machines is the formation of series of four-stroke internal combustion engines on the basis of a unified cylinder assembly and partially unified rod-and-piston assembly.

The number of possible cylinder combinations is limited because of the need to have balanced inertia forces due to reciprocating masses and a regular firing. The combinations that meet these requirements are presented in Table 3.

A greater degree of unification is characteristic of the twin crankshaft engines (XIII-XVI) which have their rod-and-piston assembly and crankshafts fully unified, as well as the cylinder assembly.

Since the power output of an engine is proportional to the number of its cylinders, the illustrated series of engines enables, in principle, a family of engines to be obtained which offers a very wide

range of power outputs. For instance, if the power output of one cylinder is 100 hp, the obtainable range of the series then will be 200-3000 hp.

However, out of the large number of schemes shown in Table 3 comparatively few are employed in practice.

Engines with a limited number of cylinders (<4) are notable for their irregular torque and bad balance.

Engines with a large number of cylinders (>24) are rarely used because of the difficulty of maintenance and greater liability to troubles.

Row engines with a small angle between cylinders (VII) are inacceptable because of the difficulty of arranging the inlet and outlet pipes between the cylinders.

In the category of low- and medium-power engines (automobile, tractor and other transport engines) use is mostly made of schemes II, III, IV and V, while in that of high-power engines (ship engines)—schemes IX, XI, and more rarely, XXIII and XXIV.

The radial schemes (XVII-XXII) were extensively used for air-cooled aero-engines.

Another application field for the method of unified machines series is represented by rotary machine tools. Since the productivity of rotary machines is directly proportional to the number of operational units installed thereon, it is possible to form a series of machines of different productive capacities by using various combinations of unified operational units. In contrast to piston engines the number of operational units which can be installed on a rotary machine is practically limitless and depends only on the required productive capacity of the machine.

Besides changing the number of the operating units on rotary machines, it is also possible to change the units themselves, thus adapting the machines for performing various functions. This is an example of combining the unified series method with conversion or multi-station machine system.

Limitation of the method. The methods of developing machines derivatives on the basis of unification cannot be regarded as universal and fully comprehensive. Each of these methods is applicable only to a limited category of machines. Many machines (steam and gas turbines) do not permit the development of machines derivatives because of their design features. Also, it is not possible or advisable to develop derivative series for special-purpose machines, high-power machines, etc., which remain in the category of individually projected devices.

Not uncommonly unification worsens the product quality, particularly in the case of derivative series having extensive ranges. The extreme members of these series are generally inferior to specially made machines as to their size, specific metalwork weight, specific

gravity and operational characteristics. Such an impairment is tolerable when unification provides for a greater economy and the size and weight are only of secondary importance.

The method is readily applicable to general-purpose machines, but has a limited application, and often inapplicable at all, to machines whose overall dimensions and weight must be kept to a minimum. In the category of high-class machines one often has to refuse unification and follow the path of individual design.

In this connection it is necessary to say a few words about the technological trend in designing wherein emphasis is placed upon processing aspects and particular attention is paid to the methods of unification and development of machines derivatives series, which are considered almost the main principle of rational design.

The chief merit of this trend is that it establishes close links between design and production. The suitability of a design for industrial production should be achieved not by a series of subsequent amendments, but through planned designing.

However, the suitability of the design for industrial production cannot be the main aim of designing.

The main trend in the machine design is the enhancement of the quality, reliability, durability and efficiency of machines. Production engineering must use all its means to help solve all these basic problems, but not determine the design trend.

Also, one must not overestimate the role of the development of machines derivatives and their series as a means for reducing the cost of machines. These methods have a limited application and their efficiency is inferior to that of other methods such as the automation and mechanization of production, reduction of the number of type-sizes of machines, etc.

It is wrong to consider the suitability of a machine for the formation on its basis of machines derivatives and their series to be an indication of the suitability of its design for industrial production even because this method cannot be applied to all machines. It would be very strange to regard, for instance, the design of a huge heat machine (e.g., a powerful steam turbine) as being unsuitable for industrial production on the grounds that no series of machines derivatives can be built up on the basis of it.

1.7. Reduction of Product Range

Decreasing the range of products on the basis of a rational selection of their types facilitates their serial manufacture, mechanization and automation of the production and introduction of progressive techniques, thus increasing the products output, cutting down their cost and improving their quality. In this way the scattering of capital in producing machines in small lots is eliminated, the

operation, maintenance and repairs of the machines are made easier, and good opportunities for a profitable manufacture of spare parts are provided.

The problem of decreasing the range of products is solved in the following three ways:

by developing parametric series of machines with rationally chosen design parameter intervals between the adjacent machines in the series;

by improving the versatility of machines, i.e., by widening their operational range;

by providing the machine designs with reserves for further development and subsequently using these reserves as the demands of the national economy grow higher.

All these methods can be combined one with another and with unification methods as well. For instance, it is possible to simultaneously develop unified and parametric series of piston engines. In this case the unified series comprise engines using the same cylinders, but differing in the cylinder number and arrangement, whereas the parametric series include engines having the same number and arrangement of cylinders, but differing in the cylinder diameter.

(*a*) *Parametric Series*

Parametric series are series of machines of the same purpose, with their design and characteristics being regulated as well as the differences in the characteristics (gradations) between the adjacent machines in the series.

In many cases it is advisable to base a series upon a single machine type, establishing the required differences in the characteristics of the machines in the series by changing the dimensions of the original machine while preserving the geometric similarity of the machines. Such series are called size-similar or simply size series.

In other instances it is advisable to establish for each range of characteristics its own machine type with its own dimensions. Such series are called type-size series.

This can be exemplified by ship engines. For low power ratings it is advisable to employ four-stroke internal combustion engines, while for medium and high power ratings it is more advantageous to use two-stroke engines as they feature smaller dimensions and weight for the same power output, or gas-turbine engines which are capable of a still greater concentration of power.

Combined series are occasionally used: some of the machine modifications in the series are of the same type and have similar geometries, while others are developed on the basis of other machine types.

The use of different machine types (as in the case of type-size and combined series) in no way affects the efficiency of the parametric

series method since the economic advantage of such series results from the reduction of the number of models. The technological gain is a centralized and hence, highly productive machine manufacture stemming from the increase in the number of each machine model produced.

The parametric series method has its maximum effect when producing machines for mass application, for such machines generally

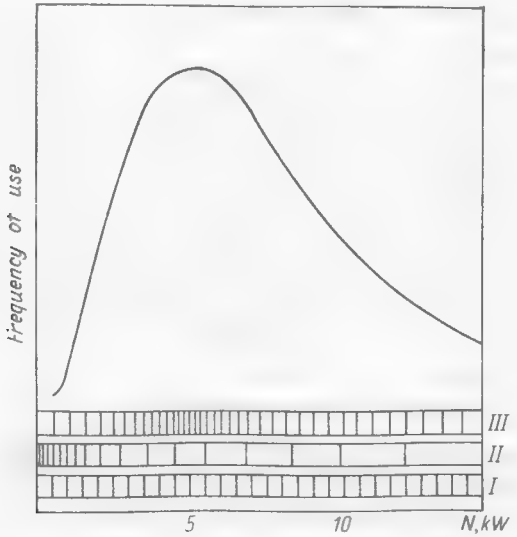


Fig. 11. Frequency of use
I — arithmetic series; *II* — geometric series; *III* — series correlated with frequency of use

have extensively ranging characteristics (internal combustion engines, electric motors, machine tools, pumps, compressors, speed reducers, etc.).

The most important factor when developing a parametric series is a correct selection of the machine types, the number of the members in the series, and the intervals between the members. When deciding upon these matters it is necessary to consider the frequency of use of various members within the series, operating conditions likely to be met with in actual service, the flexibility and adaptability of the machines of the given class (i.e., feasibility of varying their operational parameters), and possibilities for their modification and developing additional machines derivatives on their basis.

It is advisable to increase the number of the members of the series in the range of the most frequently used parameters and have larger intervals between the members in the range of parameters which are seldom required.

Let us consider the case of three-phase electric motors. Let the curve expressing the frequency of use of these motors be such as shown in Fig. 11. The scales in the lower part of the graph show schematically the motor power rating gradations obtained when developing parametric series of the motors to an arithmetic (I) and geometric (II) progressions. It is quite obvious that both these series do not accord with the curve of the frequency of use. The density of the distribution of members in the arithmetic series is the same both in the range of low and high frequency of use, which is clearly irrational. The density of the distribution of members in the geometric series is too high in the range of small power ratings and too low in the range of the maximum frequency of use.

A good compromise is presented by a rationally developed series (III). Here the members are arranged more densely in the region of the highest frequency of use and more sparsely in the region of the lowest frequency of use. Such a distribution makes it possible to satisfy more fully the requirements of the majority of consumers.

To realize fully the economic effect of parametric series, it is necessary that they be in use for a sufficiently long period of time. Therefore when developing a parametric series of machines account must be taken not only of the contemporaneous conditions, but also of the prospects of the development of the branches of industry for which the series is intended.

(*b*) *Size-Similar Series*

The designing of size-similar machines has its own particular features, the main one being that the output parameters of such machines depend not only on their geometrical dimensions, but also on the parameters of their working processes.

To maintain full similarity of the machines of different dimensions it is necessary to ensure, firstly, their geometrical similitude and, secondly, the similarity of their working processes, i.e., to make the thermal and mechanical stresses developing in the machines as a whole and in their individual parts equal for all the machines in a given series.

Similarity criteria are formulated for most machines and working processes. For instance, internal combustion engines (Fig. 12) have the following two similarity criteria:

(1) equality of mean effective pressure p_e depending on the pressure and temperature of the mixture at the inlet;

(2) equality of mean piston speed $v_p = \frac{sn}{30}$ (s — piston stroke, n—engine revolutions per minute), or equality of the product

$D \cdot n$ (D—cylinder diameter which in geometrically similar engines is related to the piston stroke by the relation $\frac{s}{D}$=const).

In a general form

$$f(p_e,\ Dh) = \text{const} \tag{1.21}$$

Should this criterion be the same, then all the geometrically similar engines will have the following identical parameters: thermodynamic efficiency, mechanical efficiency, actual efficiency (hence, the

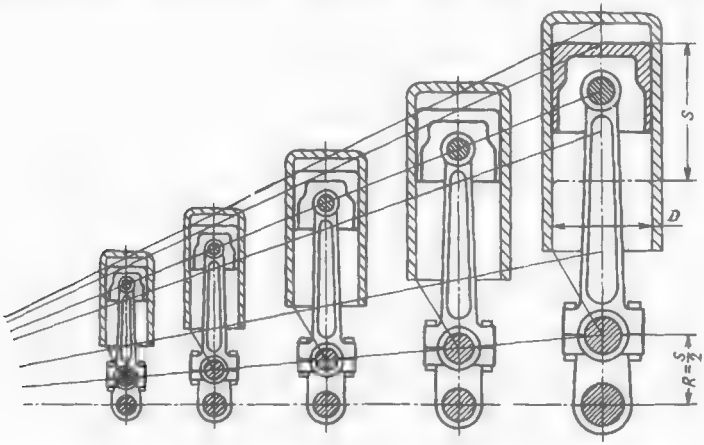


Fig. 12. Size-similar series of internal combustion engines

same specific fuel consumption in terms of g/e.h.p·h), thermal load (heat transfer per unit cooling surface), specific power, stresses due to gas pressure and inertia forces, specific bearing loads and constructional weight (weight related to the sum of the squares of the cylinder diameters).

It is clear from Eq. (1.21) that to ensure the constancy of the above listed parameters whenever the cylinder diameter is increased it is necessary to reduce either the engine revolutions per minute or the mean effective pressure. Therefore, the effective engine power is proportional to the square and not to the cube of the cylinder diameter. The litre power (power referred to the swept volume) decreases proportionally to the cylinder diameter, but the engine weight per effective horsepower rises in direct proportion to the cylinder diameter. The greater the cylinder diameter, the weaker the resistance to bending of the engine parts and the engine as a whole.

It is necessary to say that strict adherence to geometrical similitude in the region of small cylinder diameters is impracticable because of manufacturing difficulties. The minimum cross-sectional di-

mensions of the engine components are limited because of the need to make them sufficiently rigid to withstand machining (e.g., turning), assembly and transportation. This is why quite a large number of components of small-size machines are made larger than required if the conditions of geometric similitude are met. Therefore, small-cylinder engines have a greater weight per effective horsepower, but are more reliable, strong and rigid, and are capable of being supercharged and operated at increased speeds.

The above illustrated example of internal combustion engines is a particular case of an extensive category of machines in which the stresses developing in the parts depend upon the operating pressures and speeds. A general law for the machines of this class may be formulated as follows: the stresses developing in geometrically similar constructions working under similar pressures and speeds are identical.

From the above one can draw the following conclusions.

Size-similar series should be developed on the basis of the output characteristics (power, productivity, etc.) of the machines and not on their geometrical parameters (displacement, cylinder diameter, size of rotors in rotary machines) because, due to the inherent laws of similarity, the output characteristics follow a regularity which differs from that of the change of the geometrical characteristics; the latter are obtained as derivatives.

It is necessary to consider the changes in specific parameters (e.g., weight per horsepower, litre power) and mechanical properties (e.g., flexural rigidity) which are inevitable in geometrically similar machines.

(*c*) *Universalizing of Machines*

The universalizing of machines is aimed at expanding their functional potentialities and operational range, and widening the range of components machined with them. It improves the adaptability of the machines to the production requirements and increases their use factor. The chief economic gain from universalizing is that it enables the number of manufacturing units to be reduced: one universal machine is equivalent to a number of specialized machines which perform specific operations.

Widening the functions and application areas of machines can be attained by the following methods: introducing additional operating members and replacement equipment, providing built-in adjustments with a view to widening the range of the products machined, and controlling the output characteristics (speed, power, productivity).

Universalizing may be illustrated by plano-milling machines, which combine planing and milling operations, or by slabbing-bloo-

ming mills designed for turning out both blooms (billets for rolling section steel) and slabs (billets for rolling sheet steel).

Many agricultural machines are readily adaptable to universalizing. By providing a basic machine with auxiliary equipment, trailed or mounted, a multi-function machine with an extended seasonal employment interval can be readily obtained.

Universalizing techniques may be illustrated by using as an example automatic rotary filling machines intended for filling up containers of various capacities.

The first condition for universalizing a piston-type filling machine is the provision of a metering mechanism which can meter out batches within broad limits. Such a mechanism can be made in the form of a swash plate whose inclination can be changed by means of an adjusting screw. On the turnplate of the machine there are several metering cylinders with their piston rods sliding over the plate as the turntable rotates. The inclination of the swash plate determines the length of the piston stroke and, consequently, the size of the batch being metered out. The adjustment of the swash plate inclination is stepless.

The problem of handling containers of various diameters is solved by using adjustable container guideways or changeable container-feed mechanisms which deliver empty containers to the turntable for filling and then remove the filled ones. For handling containers of different heights, either the vertical position of the turnplate with the metering cylinders is changed, or that of the container-carrying turntable.

The time needed to fill a container is proportional to its capacity. Therefore, a gearbox, or a stepless speed variator is added to the filling machine to adjust the turntable rotary speed.

It is important to set reasonable limits to the degree of universality. Universal machines designed to suit too extensive a list of products or operational range are complex, heavy, cumbersome and inconvenient in use. It is sometimes more advisable to design a series of moderately universalized machines so that the complete series will provide for the required degree of universality.

In other cases universal machines may be supplemented with two or three special machines designed for parts sharply differing in size or configuration from the basic component type.

(*d*) *Consecutive Machine Development*

The resources for future development, incorporated into a machine's design, allow the machine to be systematically improved so as to keep its characteristics in line with the ever-growing demands of the national economy. Such a development method eliminates the periodic replacement of obsolete machines, assures a stable production of

machines of the same basic design over many years, and, being one of the main ways of lowering the cost of machine products, it yields a great economic gain.

The resources embodied in the design of a machine depend on the purpose of the latter. Thus, in heat engines the original model must possess ample working volume (displacement) reserves and allow increases in rotational speeds and improvements in the thermal processes to be effected. Machine tools, for which productive capacity is of prime importance, must permit increasing their speed and range of operations.

In all instances the initial model must have adequate strength and rigidity margins. This, however, does not mean that the basic model must be overweight. It is important to make stronger the most heavily stressed parts and units which may become deterrents to the intensification of the machine operation.

Immensely important is the efficiency of the kinematic scheme of the machine which predetermines the general capabilities of intensified operation inherent in the machine's design.

Not uncommonly the improvement of machines involves subsequent introduction of additional units (speed reducers, gearboxes, automation means). Their installation should not interfere with the basic design of the machine, therefore the latter must be provided with already machined seating surfaces and fastening points.

Along with the use of the original resources, the machine should be continuously improved by utilizing newly developed production and design techniques with a view to lowering the machine weight, increasing its power capacity and degree of automation, and improving its durability, reliability and maintainability.

A striking example of the above trend is the history of the Soviet aircraft engine model AM-34 which lasted for 15 years in all and, thanks to the continuous development, remained in each of its stages the best in the world of aviation. During this period its power was raised from 800 to 1800 hp through supercharging, increasing the speed and using a high-octane antiknock fuel. Engine service life rose from 200 to 1000 h. Thanks to the improvement, the engine of the last model maintained its power rating at high altitudes (up to 6000 m). The efficiency of the propeller unit was increased by using a speed reducer and a variable-pitch propeller. Though the total weight of the engine had somewhat increased (because of the additional built-in units, namely, the supercharger and reducer), its weight per horsepower was almost halved (from 0.9 to 0.5 kg/e.h.p.).

This progress was made owing to the built-in displacement reserves of the original model and systematic development of the engine without changing the basic design and initial geometry.

Machine tools may serve as another example of the advisability of providing the initial design with future development reserves.

Machine tools with increased strength, rigidity and resistance to vibration proved to be capable of being used (without alterations) for high-speed high-force cutting, while those of lower rigidity had to be reconstructed to suit the new, heavier operating conditions.

It should be emphasized that unlike other methods of cutting down the cost of machine products, which have been described above, the method of consecutive machine development is universal and applicable to all categories and classes of machines, including unique machines.

1.8. Preferred Numbers and Their Use in Designing

Standardization in mechanical engineering is based on series of numbers obeying definite regularities. Up to recent years broad use has been made of arithmetic series in which each term, after the first, is derived by adding to the preceding one a constant quantity (the common difference, or constant). With the common difference being equal to 10, an arithmetic series of diameters (from 10 to 200 mm), most popular in mechanical engineering, is as follows

10, 20, 30, 40, 50, 60, 70, 80, 90, 100, 110, 120,
130, 140, 150, 160, 170, 180, 190, 200

Arithmetic series are noted for their relative non-uniformity. Their higher regions are more densely filled with dimensional gradations than the lower ones.

The ratio of each member of an arithmetic series to the preceding one is

$$\varphi = 1 + \frac{A}{n}$$

where A = common difference (in the case under consideration $A = 10$);

n = numerical value of the preceding term

This ratio sharply decreases with the growth of the number of terms in the series. Thus, in the above series the ratio of the second term to the first one is $\varphi = 2$; fifth to fourth, $\varphi = 1.25$; tenth to ninth, $\varphi = 1.1$; twentieth to nineteenth, $\varphi = 1.05$.

The nonuniformity may be partially corrected by changing the value of A for different regions of the series. Thus, for the case under consideration, within the regions $D < 50$, $D = 50\text{-}100$ and $D > 100$, A can be taken at 5, 10 and 20, respectively.

As a result, the following series is obtained

10, 15, 20, 25, 30, 35, 40, 45, 50, 60, 70, 80, 90, 100,
120, 140, 160, 180, 200

This series is more uniform, though with a stepwise change of the size gradations.

More rational are series developed on the principle of the geometric progression in which each term, after the first, is derived by multiplying the preceding one by a constant value φ (the constant ratio).

(a) *Basic Preferred Numbers Series*

The USSR State Standard ГОСТ 8032-56 stipulates four series of preferred numbers with different values of

$$\varphi = \sqrt[n]{\frac{l}{a}} \tag{1.22}$$

where a and l are the first and the last terms of a series, respectively. In the standard series $\frac{l}{a} = 10$. Then, Eq. (1.22) becomes

$$\varphi = \sqrt[n]{10}$$

The index n of the radical is taken at 5, 10, 20 and 40. These figures preceded by the letter R make up the designation of a series.

In this way series $R5$, $R10$, $R20$ and $R40$ are obtained, for which the value of φ is $\sqrt[5]{10} \approx 1.6$, $\sqrt[10]{10} \approx 1.25$, $\sqrt[20]{10} \approx 1.12$ and $\sqrt[40]{10} \approx$ ≈ 1.06, respectively.

Any term in a series is

$$a_k = a\varphi^k$$

where k = serial number of the term;

a = the first term of the series, to which the zero serial number is assigned

With a decrease in the value of φ the intervals between the terms in a series become shorter, the total number of the terms grows larger, and the series itself becomes more fractional. As an exception, the use of the very fractional series $R\,80$ with $\varphi = \sqrt[80]{10} \approx 1.03$ is permissible.

According to the USSR State Standard ГОСТ 8032-56 the basic series of preferred numbers in the range of 1-10 are as follows

R5: 1; 1.6; 2.5; 4; 6.3; 10

R10: 1; 1.25; 1.6; 2; 2.5; 3.15; 4; 5; 6.3; 8; 10

R20: 1; 1.12; 1.25; 1.4; 1.6; 1.8; 2; 2.24; 2.5; 2.8; 3.15; 4; 4.5; 5; 5.6; 6.3; 7.1; 8; 9; 10

R40: 1; 1.06; 1.12; 1.18; 1.25; 1.32; 1.4; 1.5; 1.6; 1.7; 1.8; 1.9; 2; 2.12; 2.24; 2.36; 2.5; 2.65; 2.8; 3; 3.15; 3.35; 3.55; 4; 4.25; 4.5; 4.75; 5; 5.3; 5.6; 6; 6.3; 6.7; 7.1; 7.5; 8; 8.5; 9; 9.5; 10

The numerical values of the terms of all series are rounded off to within ±1%. Each subsequent lower series is obtained by deleting even terms from the preceding higher series.

An additional series with the common ratio $\varphi = 1.03$ is

***R*80: 1; 1.03; 1.06; 1.12; 1.15; etc.**

(*b*) *Derived Series* (*Subseries*)

From the basic series one can obtain geometric series for any range of numbers, i.e., with any value of the first and the last terms. In accordance with the basic law governing the development of geometric progressions, derived series are obtained by multiplying successively the first term of a new series by the terms of any basic series (*R*5, *R*10, etc.) until the quantity $10a$ is obtained, which, in its turn, is multiplied by the terms of the same basic series, etc.

Let us cite as an example a series of numbers ranging from 1 to 1000, derived on the basis of the *R*5 series

1; 1.6; 2.5; 4; 6.3; 10; 16; 25; 40; 63; 100; 160; 250;
400; 630; 1000

Series developed on the basis of geometric progression can be made sparser by selecting the *m*th terms (*m*—a serial number multiple of any integer). As a result, a new series with a common ratio φ^m is obtained. Examples of such a thinning out are, in fact, the basic series of preferred numbers.

The series *R*20 ($\varphi^m = 1.06^2 = 1.12$), *R*10 ($\varphi^m = 1.06^4 = 1.25$) and *R*5 ($\varphi^m = 1.06^8 = 1.6$) are obtained by selecting from the *R*40 series ($\varphi = 1.06$) all the terms whose serial numbers are multiples of 2, 4 and 8, respectively. By selecting from the *R*40 series the terms whose serial numbers are multiples of 3, 6 and 9 it is possible to obtain accordingly series with the following common ratios

$$\varphi^m = 1.06^3 = 1.19; \quad \varphi^m = 1.06^6 = 1.41; \quad \text{and} \quad \varphi^m = 1.06^9 = 1.68$$

Derived series can also be obtained by other methods. Raising the terms of a geometric progression to any power gives a new progression with a new common ratio. Thus, when terms of the *R*5 series are squared, the following progression with a common ratio of 2.56 is obtained

1; 2.56; 6.25; 16; 39.7; 1000

Thus, if the linear dimensions of a series of parts from a geometric progression, the magnitudes of their cross sections, volumes, weights, resisting moments, and moments of inertia of sections then also form geometric progressions, but with other common ratios any different first and last terms.

From the ability of the cross sections, resisting moments and moments of inertia to form geometric progressions it cannot be concluded that all the parts whose linear dimensions follow the order of a geometric progression display the properties of equal strength and equal rigidity. For this to be true the acting loads should be proportional to the square of the linear dimensions of the parts, a case very seldom met with in practice.

(c) *Standard Linear Dimensions*

On the grounds of the basic series of preferred numbers there have been developed series of standard linear dimensions (USSR State Standard ГОСТ 6636-60) differing from the basic series in that their numbers are rounded off more roughly. The series of standard linear dimensions are designated as *R*a5, *R*a10, *R*a20, and *R*a40.

***Ra*5:** 0.1; 0.16; 0.25; 0.4; 0.6; 1; 1.16; 2.5; 4; 6; 10; 16; 25; 40; 60

***Ra*10:** 0.1; 0.12; 0.16; 0.2; 0.25; 0.32; 0.4; 0.5; 0.6; 0.8; 1; 1.2; 1.6; 2; 2.5; 3; 4; 5; 6; 8; 10; 12; 16; 20; 25; 32; 40; 50; 60; 80

***Ra*20:** 0.1; 0.11; 0.12; 0.14; 0.16; 0.18; 0.2; 0.22; 0.25; 0.28; 0.32; 0.36; 0.4; 0.45; 0.5; 0.55; 0.6; 0.7; 0.8; 0.9; 1; 1.1; 1.2; 1.4; 1.6; 1.8; 2; 2.2; 2.5; 2.8; 3; 3.6; 4; 4.5; 5; 5.5; 6; 7; 8; 9; 10; 11; 12; 14; 16; 18; 20; 22; 25; 28; 32; 36; 40; 45; 50; 55; 60; 70; 80; 90

***Ra*40:** 0.1; 0.105; 0.11; 0.115; 0.12; 0.13; 0.14; 0.15; 0.16; 0.17; 0.18; 0.19; 0.20; 0.21; 0.22; 0.24; 0.25; 0.26; 0.28; 0.3; 0.32; 0.34; 0.36; 0.38; 0.4; 0.42; 0.45; 0.48; 0.5; 0.52; 0.55; 0.6; 0.63; 0.65; 0.7; 0.75; 0.8; 0.85; 0.9; 0.95; 1; 1.05; 1.1; 1.15; 1.2; 1.3; 1.4; 1.5; 1.6; 1.7; 1.8; 1.9; 2; 2.1; 2.2; 2.4; 2.5; 2.6; 2.8; 3; 3.2; 3.4; 3.6; 3.8; 4; 4.2; 4.5; 4.8; 5; 5.2; 6; 6.3; 6.5; 7; 7.5; 8; 8.5; 9; 9.5; 10; 10.5; 11; 11.5; 12; 13; 14; 15; 16; 17; 18; 19; 20; 21; 22; 24; 25; 26; 28; 30; 32; 34; 36; 38; 40; 42; 45; 48; 50; 52; 55; 60; 63; 65; 70; 75; 80; 85; 90; 95

The ГОСТ 6636-60 Standard covers dimensions up to 95 mm. When necessary, these series can be continued by following the principle of geometric progression while keeping the common ratio unchanged. For instance, for the range of 100-250 we have

***Ra*5:** 100; 160; 250

***Ra*10:** 100; 120; 160; 200; 250

***Ra*20:** 100; 110; 120; 140; 160; 180; 200; 220; 250

***Ra*40:** 100; 105; 110; 115; 120; 130; 140; 150; 160; 170; 180; 190; 200; 210; 220; 240; 250

The application of the standard linear dimensions is advisable for surfaces subject to an accurate machining, especially for connection joints*. This helps standardize cutting, checking and measuring tools, and facilitates the setting up of machine tools.

The greatest economic gain is obtained when reducing the number of the terms in a series, i.e., when using in each individual case the lowest series providing for the necessary size range.

Standard dimensions which are referred to surfaces not involving precise co-ordination are of less importance.

The series of the diameters of wire and bar-steel rolled stock, thickness of sheet-steel rolled stock, linear cross-sectional dimensions of shape-steel rolled stock—all are based on the standard linear dimensions.

The series of round rolled stock must be consistent with the series of machined diameters, so that adequate machining allowances are given, providing for the minimum waste of metal in the form of chips.

If the *Ra*10 series containing dimensions 10, 12, 16, 20, 25, . . ., is taken as the basic one for machining purposes, then for the blanks it is more advisable to take a modified *Ra*20 series obtained by selecting from the latter series only those dimensions which are shifted by one serial number relative to the dimensions of the basic *Ra*10 series. Thus, one may obtain the following series:

Diameters of machined workpieces, mm	10	12	16	20	25	32	40	50	60	80
Diameters of blanks, mm	11	14	18	22	28	36	45	55	70	90

The setting up of the diameters series for blanks and machined workpieces by following the same principle and with the same common ratio can lead to a greater waste of metal in the form of chips.

It is irrational to use the series of preferred dimensions for surfaces not subject to machining (castings, forgings). In such instances even a partial standardization of dimensions has no actual advantages and only makes the process of design and manufacture more difficult.

Series of preferred numbers used in designing. The significance of preferred numbers used in designing should not be overestimated. Some authors are of the opinion that the series of preferred numbers must be used not only for the purposes of standardization, but also for all the spheres of designing. They are wrong.

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* The USSR State Standards pertaining to the tolerances and fits for general-purpose joints, as well as antifriction bearings, are not yet brought in line with the preferred numbers series.

The use of preferred numbers series is advisable when it is necessary to establish a sequence of gradations of some parameter, the gradations being uniformly distributed throughout the entire range of the series (e.g., gear ratios in speed and feed boxes of metal-cutting machine tools).

However, such a uniform distribution of gradations is not always the most rational. It would be more correct, in principle, to proceed from the density of distribution of the frequency of use of a given parameter.

As an example, the graph in Fig. 13 shows the frequency of use of gear modules in general mechanical engineering. It is obvious that

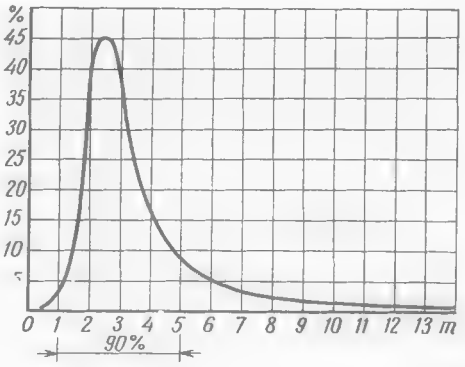


Fig. 13. Frequency of use of gear modules in general mechanichal engineering

90% of all the gears used have their modules lying within the range of m = 1-5. The maximum application find gears with modules m = 2-3. Thus, it is reasonable to increase the number of gradations in the range of the highest frequency of use and reduce it in the region of the least popular modules. In other mechanical engineering branches (instrument-making, heavy machine-building) relations may be different. In each of such branches it is possible to determine the density of distribution of the frequency of use and then select accordingly the standard module gradations. In essence, such a differentiated approach to standardization is also necessary in the case of other mechanical engineering parameters (sizes of fitting diameters, threads, etc.).

The series of preferred numbers are inapplicable when developing unified series of machines with repeatedly used working members. Parameters of unified series follow other laws, which are dependent on the feasibility of combining unified members and conditions of the technical application of the members of the series, and cannot be arranged in the order of a geometric progression.

When setting up a parametric series due account must be taken of the frequency of use of different machine categories, their flexibility, etc. Formal application of geometric progressions may lead to serious errors.

The series of preferred numbers are also inapplicable to the determination of the parameters of progressively developed and modernized machines whose characteristics at each development stage depend on the technical capabilities and requirements of the corresponding branches of industry. Thus, the power capacity of heat engines is dependent on their original parameters (pressure and temperature) and speed. Not one of these parameters can be arbitrarily increased. In some cases they have an optimum value (e.g., the degree of compression in gas turbines), whose changes worsen the machine characteristics. Increasing the temperature and speed can only be done on the basis of technical improvements (increasing the heat resistance of materials, improving the cooling of thermally stressed parts). The results of such investigations cannot be kept within preferred numbers series.

1.9. General Design Rules

When designing a machine, it is recommended to keep to the following basic rules:

design the machine with a view to enhancing the economic effect which is primarily dependent on the efficiency and durability of the machine and its maintenance costs throughout the entire service life;

achieve a maximum output by increasing the productive capacity of the machine and extending its range of operations;

take all measures to lower operational expenditures by decreasing the power consumption and costs of servicing and repairs;

automate the machine as fully as possible with the aim of increasing its productivity and the product quality and reducing labour costs;

increase the machine's durability as much as possible in order to enlarge the actual fleet of machines and raise their total output;

minimize the risks of obsolescence by endowing the machine with high initial parameters and reserves for future development and improvement;

make the machine capable of being intensively operated by improving its reliability and versatility;

provide possibilities for manufacturing derivatives on the basis of the original model;

reduce the number of the type-sizes of machines as much as possible and improve their operational flexibility;

strive to satisfy the needs of the national economy not by producing more machines but by increasing their output and durability;

design the machine so as to assure its repairless operation with the complete exclusion of capital repairs and reduction of all other repairs to unit replacement;

avoid the design of rubbing surfaces directly on housings or machine bodies; to facilitate repairs of such rubbing surfaces design them on individual and easily changeable parts;

follow consistently the unit design principle (multi-station machine system); construct machine assemblies as self-contained units;

exclude special selection and matching of parts during their assembly; make them fully interchangeable;

exclude the alignment and adjustment of parts and units during assembly; provide the necessary locating elements that enable correct mounting of the parts and units;

impart high strength to machine parts and to the machine as a whole without increasing their weight (use the most rational shapes providing for the best utilization of the properties of materials, employ high-strength materials, introduce hardening treatment, etc.);

pay particular attention to increasing the cyclic strengths of parts; use shapes providing for a high fatigue strength; eliminate stress concentrations; introduce treatment improving the fatigue strength (induction hardening, nitriding, shot-blast work hardening, etc.);

in machines, units and mechanisms, operating under cyclic or dynamic loads, introduce resilient elements which dampen impacts and load fluctuations;

make constructions highly rigid without increasing their weight (use hollow and shell-type constructions; avoid deformation with the aid of diagonal and transverse braces; select rational positions for supports and stiffening elements);

increase operational reliability in every way possible, thus making the operation of the machine wholly trouble-free;

make machines as simple to maintain as possible; reduce the scope of maintenance; eliminate periodic adjustments; make mechanisms as self-servicing units;

prevent the overloading of the machine during operation; introduce automatic monitors, protection and limiting devices that should prevent the machine from working under dangerous conditions;

exclude failures and accidents ensuing from unskilled and careless attendance; introduce interlocking devices which should exclude wrong manipulation of controls; make the machine control as automatic as possible;

exclude any possibility for an incorrect assembly of machine parts and units requiring precise coordination with respect to one another; introduce interlocks which should only permit correct assembly;

eliminate periodic lubrication; ensure an automatic and continuous feed of the lubricant to rubbing surfaces;

avoid open mechanisms and devices; enclose all mechanisms in housings which should prevent dust, dirt and moisture from getting onto rubbing surfaces and allow a continuous lubrication to be effected;

assure a reliable locking of screwed connections to prevent them from self-loosening, lock internal connections by positive methods (apply cotters, keys, tab washers, etc.);

prevent corrosion of parts, particularly in machines operating in the open air or coming into close contact with chemically aggressive media, by applying chemically stable coatings, in the form of either paints and varnishes or electroplates, and also by manufacturing the parts of corrosion-resistant materials;

cut down the cost of the machine by making its design suitable for industrial production, and also through unification, standardization, decreasing the specific metalwork weight and reducing the number of the machine type-sizes;

reduce the weight of the machine by making it more compact and also by applying rational kinematic and loading schemes, eliminating unfavourable kinds of loading, changing bending stresses to tensile or compressive stresses and using light alloys and non-metallic materials;

make the machine design as simple as possible; avoid complex multi-component constructions;

wherever possible change mechanisms with a rectilinear reciprocal motion to the more advantageous rotary mechanisms;

assure the maximum suitability of parts, units and the machine as a whole for industrial production by building into its design preconditions for the most efficient manufacture and assembly;

reduce machining operations; anticipate the manufacture of parts from blanks whose shape is close to that of the finished parts; change the machining to other more productive methods which do not involve metal cutting;

implement the maximum unification of constructional elements with a view to reducing the cost of the machine, shortening the time of its manufacture and facilitating its operation and repairs;

broaden the use of standard parts; adhere to the national and local standards in force;

never use specially designed parts or units when it is possible to employ standard, unified, adopted and purchased parts and units;

economize on expensive materials and materials in short supply by replacing them with equally good substitutes; if the use of the materials in short supply cannot be avoided, keep their consumption to the minimum;

cheapen the production but do not limit expenses on the manufacture of vital parts determining the machine's durability and reliability; produce such parts from high-grade materials and use modern

production processes providing for increased reliability and service life;

impart to the machine simple and smooth shapes which should facilitate keeping it in neat and tidy condition;

pay proper attention to the industrial design requirements, make the machine well-proportioned and provide good external finishes;

wherever possible concentrate controls in a single place convenent for manipulation and observation;

make the machine units and mechanism which require periodic inspection easily accessible;

ensure safety of attending personnel; preclude accidents by making the operation of the machine as fully automatic as possible, by introducing interlocks, enclosing mechanisms and by providing guards;

in machine-tools and automatic machines provide devices that should enable the adjustment and setting-up to be carried out while driving the machine manually; provide the possibility for the machine to be slowly driven from the drive motor (with reverse, when required for the machine setting-up);

in machines having electric motor drives consider the possibilities of wrong switchings, while in those driven from internal combustion engines take precautions to avoid the back-firing of the engine; provide reversible drives or introduce safety devices (e.g., overrunning clutches);

carefully study the experience gained in running the machines; eliminate without delay all defects discovered during use; remember, that studies into the operation of machines are the best means of improving and refining them and an effective method of improving the skill of the designer;

continuously improve the design of machines which are being produced on a mass scale so as to keep them in line with the ever-growing industrial demands;

anticipate new designs, thus preparing in advance for the production of new, more perfect machines to replace obsolete models;

when developing new designs, as well as machines intended for new technological processes, check all new elements by experiment, simulated processes, test runs, etc.;

make wider use of known designs; use the experience gained in the allied and, if necessary, more remote branches of mechanical engineering.

Chapter 2

Design methods

The starting material for designing may be the following:

a technical assignment worked out by a planning organization or customer; such an assignment generally specifies the required parameters of the machine, and the field and conditions of its use;

a technical suggestion initiated by a designing organization or a group of designers;

a research work or an experimental model based on the research;

an invention proposition or an invention prototype;

a machine prototype of foreign make which has to be reproduced with modifications and alterations.

The first case is more general and most convenient to following the design processes.

It is necessary to approach any technical assignment critically. The designer must well know the branch of industry for which the machine is intended. He must verify the assignment and, when necessary, prove the need for corrections.

A critical approach is particularly necessary when the customer is an individual factory or branch of industry. In the latter case the designer must not only satisfy the customer's requirements but also consider that it is possible for the machine to be used at other factories and in the allied branches of industry.

A circumstance not always considered is the fact that from the moment of the inception of designing to the introduction of the machine into production a time period elapses, which is the longer the more complex the machine. This period includes the following stages: design, manufacture, finishing and adjustment of the prototype, industrial testing, introduction of necessary alterations to eliminate defects found when testing, State Acceptance Tests and, finally, acceptance of the prototype. This is followed by the preparation of technical documents for the initial production series, its manufacture, and industrial tests, after which documents for mass production are prepared, the production processes modified to suit mass production, and then the production is started.

Under the most favourable conditions when there are no large troubles or complications this process takes about a year and a half or two years. Sometimes from the beginning of designing to the beginning of mass production two or three, or even more years elapse. With the modern tempo of the technological progress in the mechanical engineering this seems a long time.

Machines with incorrectly chosen, underestimated parameters, based on trite solutions, not assuring technical progress and incompatible with new ideas about the import of quality, reliability and durability, become obsolete even before the initiation of mass production. Then, the whole time spent on designing, manufacture and tests will be vain and the industry will still lack the machine it needs.

2.1. Design Succession

Succession in design implies the use during designing of the previous experience gained in the given branch of mechanical engineering, as well as in the allied branches, and introduction into the projected unit all the useful features of the existing machines*.

Nearly every modern machine is the result of the work of several generations of designers. The original model of a machine is gradually improved, equipped with new units and mechanisms, and made more perfect as a result of new constructional solutions, fruits of the creative efforts of many people. Some design solutions die due to the development of more rational ones, new technological procedures or higher operational demands, while some, proved exceptional, remain in force for very long periods, although at times being slightly modified.

In the course of time, machine specifications, both technical and economic, are made higher, their power and productive capacities are enhanced, automation is more extensively introduced, reliability and durability are bettered. Alongside the improvements to existing machines new machines of identical purpose are devised, which are principally based on other design schemes. In competition more progressive and viable designs win.

When reviewing the history of mechanical engineering it is seen that a huge number of miscellaneous design schemes were tried before. Many of them disappeared and became completely forgotten, were revived after tens of years on a new technological basis, and again returned to use. Thus historical reviews help the designer to avoid errors and the repetitions of the stages traversed, and allow him to choose more prospective methods of machine development.

* Some authors use the design succession concept to denote the general tendency of typification, unification, development of standard series, etc. Such use of the term does not correspond to the sense of its content.

It is useful to plot graphs showing the changes of basic machine parameters over the years. Tendencies of constructional forms are well expressed by graphs showing as percentages the occurrence of various designs. An analysis of such graphs and their extrapolation make it possible for a clear picture to be drawn of future machine parameters and design trends.

Of particular importance is the study of the starting data when working on a new design. Here the main task is to choose machine parameters correctly. Partial design errors can be eliminated during machine manufacture and tests, but errors in basic parameters are often irrepairable and may lead to failure of the design. Therefore, at this stage neither time nor effort on investigations should be spared. Here, more than anywhere, the proverb "look before you leap" is valid.

The selection of parameters must be preceded by a comprehensive study of all factors determining the machine's life. It is necessary to carefully study the experience gained in the operation of foreign and domestic machines, correctly analyse their advantages and disadvantages, choose a correct prototype, and make clear the tendencies of the development and requirements of the branch of engineering for which the machine is intended.

An important prerequisite for correct design is the availability of pertinent references on constructional materials. Apart from archives of their own products, design organizations should have at their disposal design catalogues from neighbouring organizations.

It is necessary to undertake systematic studies of patents and periodicals, both foreign and domestic.

Owing to the general tendency of capitalist firms to keep their novel technicalities secret, publications are often camouflaged. Therefore the designer must be able to read between the lines because sometimes a brief information may conceal an important innovation in a given branch of engineering.

In his particular sphere the designer during the course of his searches must process the scientific investigations of research institutions.

In addition to the researches carried out in its own branch, the design organization should make wide use of the experiences of other, sometimes rather alien branches of mechanical engineering. This widens the outlook of the designer and enriches his store of constructional means and devices.

It is especially useful to study the experience of advanced branches of engineering, in which the design and technological concepts, forced by the high demands for the product quality (aviation) and mass production (automotive industry), continuously develop new designs, methods of increasing strength, reliability and durability, and better production techniques.

The use of the experience already gained often helps solve partial problems arising during designing. Sometimes the designer wrestles over the development of a specialized unit or assembly new for the given machine, but which has long ago been developed and proved in other branches of engineering.

The concept of design succession does not mean limitation of initiative. The designing of any machine presents an unlimited field of activity to the designer. He must not invent anything already invented and must not forget the rule formulated early in the 20th century by Güldner: "*weniger erfinden, mehr konstruiren*" (invent less, design more).

The process of continuous machine improvement, caused by the ever-growing demands of the industry and national economy, results in the development of the design school and the cast of the design thought. The striving for a perfect design penetrates the flesh and blood of the designer, becomes his life. A genuine designer is perpetually charged with vigour to wrestle with any difficulties. He will feel fully satisfied only when his numerous relentless efforts, failures and errors are finally crowned with the finding of a most perfect solution promoting the progress in a given branch of engineering.

The designer must constantly improve and enrich his store of design solutions. An experienced designer will invariably note and mentally "snapshot" interesting solutions, even in machines "foreign" to him.

The designer must always be aware of all the latest technological processes, including physical, electrophysical and electrochemical methods of processing (spark-erosion, electron-beam, laser, ultrasonic, dimensional electrochemical etching, processing with blast, electrohydraulic impact, electromagnetic impulses, etc.). Otherwise he will be rather restricted in his choice of the most rational shapes for parts and unable to build into the design the conditions for productive manufacture.

2.2. Study of Machine Application Field

The progress in mechanical engineering is inextricably linked with the development of the different branches of industry and national economy, appearing as the machine users. Industrial development is accomplished by continuous improvements, including: growth of productivity; decrease of operational cycle; development of new technological processes; rearrangement of production lines; change in equipment layout; continuous expansion of mechanization and automation of production. This accordingly results in higher requirements for machine parameters, their productivity and level of automation. As new technological processes come into

being, some machines become obsolete. The necessity arises for new machines or for modification of the old.

Occasionally such modifications may be very extensive and affect large categories of machines. Thus, the introduction of the continuous steel casting process means the death or, in any case, reduction in the manufacture of such complex and large machines as blooming and slabbing mills. Production of steel in oxygen blast converters will reduce the use of open-hearth furnaces unless they are radically improved. The introduction of magnetogasdynamic generators, enabling thermal energy to be directly converted into electrical energy, will cause die-out of electric generators and considerable reductions in the employment of heat engines.

Before designing machines intended for a specific branch of industry, it is necessary to examine carefully the dynamics of the quantitative and qualitative development of that particular branch, the requirements for the given machine category and the possibility of the development of new technological processes and new methods of production.

The designer must be well acquainted with the particular characteristics of that branch of industry and the working conditions of the machine. The best designers, from the author's observations, are those who have been practical engineers and combine their design aptitudes with knowledge of running conditions for the machines being designed.

When choosing machine parameters it is necessary to consider actual working conditions. For instance, the productivity of a machine must not be arbitrarily increased without considering production requirements for which the machine is intended and without considering the needs of adjacent machines. Under certain conditions machines with higher productivity may operate underloaded and be more idle than at work. This lowers the degree of their use and lessens the economic advantages.

2.3. Choice of Design

When choosing the parameters of a machine, its basic scheme and type of construction, attention must be given to the factors determining its economic efficiency, i.e., high output, low energy consumption, low maintenance and operation expenditures, long service.

Generally, the final machine version is chosen from several, which have been carefully evaluated and compared from all sides of constructional expediences, perfection of kinematic and force schemes, cost of manufacture, energy consumption, cost of labour, reliable operation, size, metal content, weight, suitability for industrial production, unitizing, servicing, assembly-disassembly, inspection, setting and adjustment.

It must be made clear to what degree the original design assures future development, forced working and subsequent improvement, forming on its basis of machines derivatives and modified versions.

Not always, even during the most careful search, is a successful solution found, which fully answers the posed demands. Exceptional in constructional practice is a variant being a success in all respects. The matter is sometimes not the lack of inventiveness, but rather conflicting requirements being imposed. Under such circumstances it is necessary to find a compromise solution, waiving some requirements which are not of prime importance in the given conditions of the machine application. Often a variant is chosen not because it incorporates more advantages, but rather because it has less disadvantages in comparison with others.

After selecting the scheme and basic parameters of the machine, a sketch is made, and then a general arrangement drawing, on the basis of which are composed the initial, technical and working designs.

2.4. Development of Design Versions

The development of design versions is not a matter of individual habits or leanings of the designer, but a regular design method aimed at seeking the most rational solution.

As an example of the development and comparative analysis of design versions, let us consider a bevel reduction gear unit (Fig. 14) which is very often used in mechanical engineering.

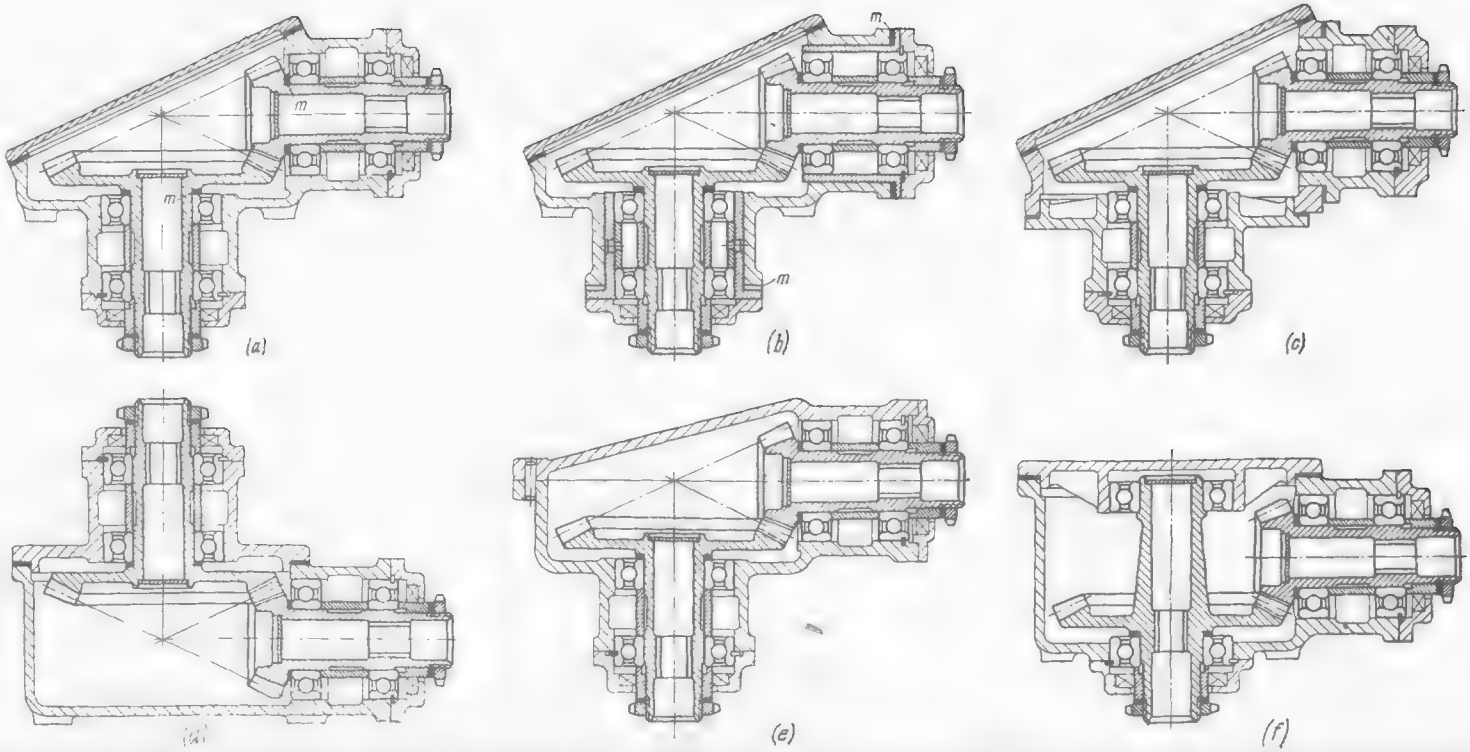
To simplify the example we will not consider constructional variants of power input and take-off, types of supports, methods of fixing the axial position of gears. We shall examine only the working arrangement of the drives, casing construction and spacing of supports.

Widely used in practice are reduction gear units which have cantilevered gears mounted inside a common casing (Fig. 14*a*). The popularity of this design is due to its undoubted advantages. Gear shafts are located inside a single casing enabling a highly accurate mutual positioning of gears to be attained during manufacture. Free access is effected through an inspection hole provided with a detachable cover. The entire mechanism can easily be inspected when assembled.

Gear mesh is controlled by spacing washers m whose installation requires the disassembly of the gear mounting unit.

Generally, the reduction gear unit is held in position by lugs. For assembly purposes the outer diameter of the pinion must be less than that of the holes holding the shaft bearings.

In another alternative (Fig. 14*b*) the shaft bearings are mounted in intermediate sleeves, thus enabling the pinion diameter, admissible from the standpoint of assembly conditions, to be slightly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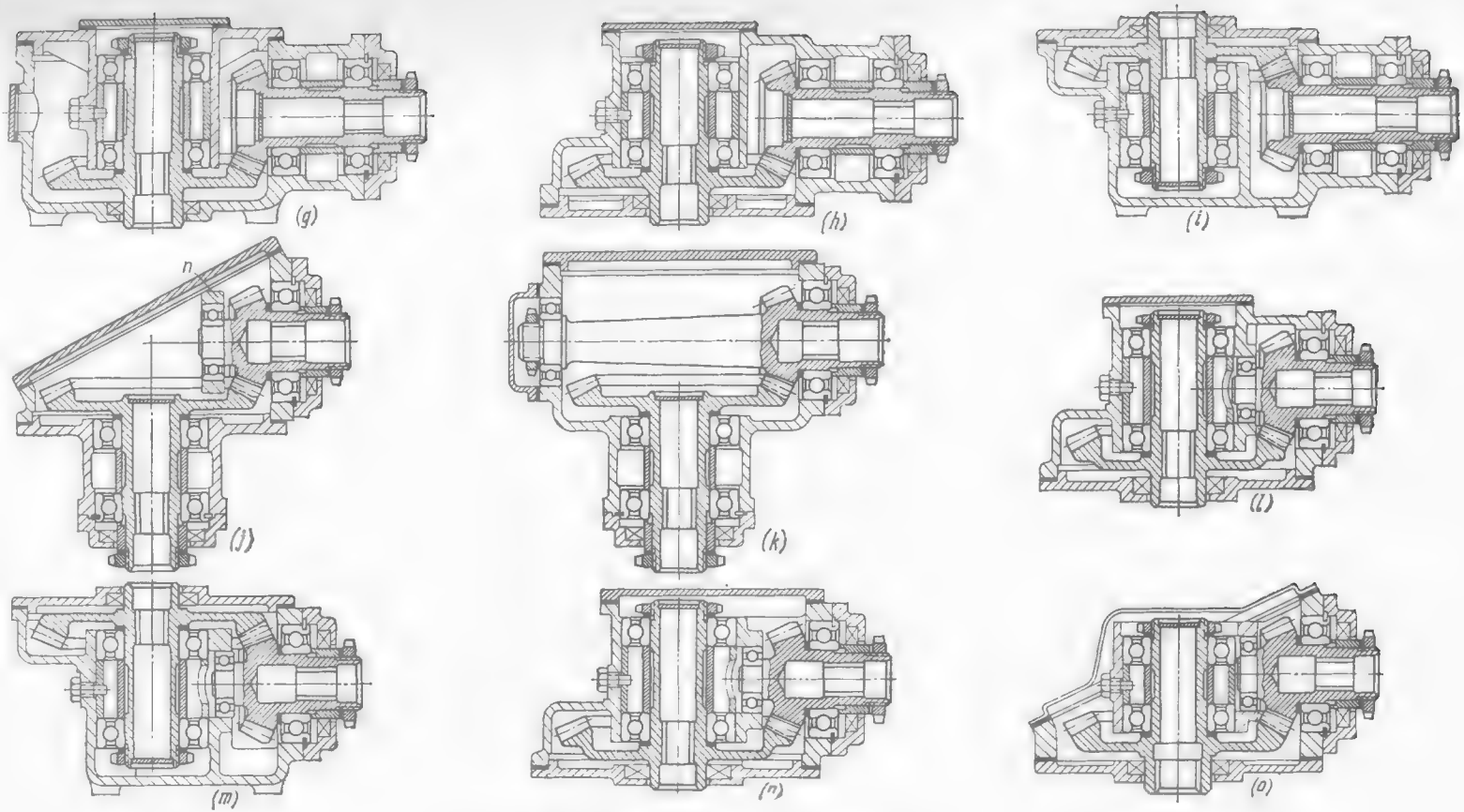


Fig. 14. Angle drive design versions

increased. This facilitates the adjustment of the mesh, as in this case it is not necessary to dismantle each gear, but merely change spacing washers *m* placed under the flanges of the intermediate sleeves. This is particularly easy if the spacer washers *m* are made as half-rings locked in place by bolts (Fig. 14*b*, lower part). To change the washers the bolts are unscrewed and the intermediate sleeve shifted to suit the new washer.

In the construction illustrated in Fig. 14*c* the gears are installed inside detachable casings. This version preserves the advantages of Fig. 14*b* but the rigidity of the main casing is worse. This means that the alignment of centering shoulders and seating surfaces of bearings must be kept to extremely close tolerances when manufacturing the casing. According to its design features this reduction unit is more suitable for suspension-type mounting, although it can also be mounted on feet (cast integral with the lower cover).

The design is worse, if the gear shank is extended upward (Fig. 14*d*). Now the mechanism cannot be inspected when assembled and the removal of the gear casing disturbs the entire unit.

Gear mesh can be controlled only by blacking-in, which means dismantling and refitting the gear many times. In order to inspect the reduction unit interior it is necessary to disconnect the gear drive shaft.

In the design shown in Fig. 14*e* the gears are installed inside a split casing (the split line runs along the pinion axis). The design is simple to assemble and easy to inspect. Inspection of gear engagement can only be complete when the pinion shaft in assembly with the bearings is pressed against the lower bearing seatings.

Machining of a split casing is more difficult. First, it is necessary to finely machine the joint surfaces, connect two halves by locating dowels and machine the bearing seats after assembly. The joint surfaces must be lapped. Joint seals cannot be used, as their use spoils the true cylindrical shape of the bearing seats.

The remaining variants have the aim of reducing overall size by bringing the supports into the casing interior.

In the design in Fig. 14*f* the gear shaft is mounted in two supports, one of which is inside the casing, and the other, in the cover. Here the distance between the supports is increased and bearing loads are less than in the cantilever design. The chief disadvantage of this construction is the difficulty of inspection and adjustment of the mechanism. When casing cover is removed, the gear shaft is held by the lower support only and because of this freedom the mesh accuracy cannot be checked. Furthermore, since the bearings are located in different casing sections, the shaft alignment is worse. The necessity of machining the bearing seats in the assembly of the casing and cover complicates the process technology.

The more compact design illustrated in Fig. 14*g* proves unsatisfactory when examined closely. Here it is practically impossible to inspect the mechanism. The removal of the gear is only possible after the cover and pinion have been dismantled, consequently, the mechanism must be fully disassembled.

A more preferable design is shown in Fig. 14*h*, in which the gear shaft is mounted inside the casing. Here the entire mechanism is accessible through the bottom cover. The design is suitable only when suspension-mounted and cannot be used if installed upon a stand by its bottom, as it would be necessary to detach the entire reduction unit in order to inspect it. The advantage of this design is the possibility of taking power from the upper end of the gear shaft for which it is sufficient to extend the shaft through the upper cover.

A similar design is given in Fig. 14*i*. To inspect the mechanism it is necessary to disconnect the gear drive. The meshing portion is viewed from the end face.

The size of the transmission can be decreased by bringing one of the pinion shaft bearings into the casing (Fig. 14*j*). The bearing is installed in partition *n* cast integral with the casing sides. This design is suitable for both suspension and installation by its base. The mechanism is inspected through the non-bearing cover; the drive shafts remain connected. The disadvantage of this design is in that the inspection of the meshing portion, screened by partition *n*, is rather difficult.

In the alternative illustrated in Fig. 14*k* the internal pinion shaft bearing is transferred to the opposite casing wall. The design features well-spaced supports and convenience of the mechanism inspection obviating the necessity to disconnect the drive shafts. Power take-off from the pinion shaft is possible.

The disadvantage: the gears cannot be dismantled separately; to dismantle the gear it is necessary first to dismantle the pinion.

From the dimensional viewpoint the most preferable designs are those in which the gear and pinion bearings are accommodated inside the casing (Fig. 14*l*, *o*).

In the design shown in Fig. 14*l* the bearings of the gear shaft are mounted in an integrally cast hub. The inside pinion bearing is accommodated in the side hole of the hub. The gear is mounted through the lower non-bearing cover, which is also an access for inspecting the mechanism. Gear mesh is viewed from the end face. This design is only suitable for suspension mounting.

Figure 14*m* shows another type of reduction unit with the output end of the gear shaft upward. The unit is mounted by its lower surface.

The disadvantage of the alternatives in Fig. 14*l*, *m* is that the gear drive shaft must be disconnected before the inspection cover can be removed.

The variant illustrated in Fig. 14*n* enables inspection without the disconnection of the drive.

Another alternative, featuring reduced casing height and a large-size moulded lid, contributing to easier inspection is shown in Fig. 14*o*.

The final choice of a reduction unit design depends on its instàllation and application. For general use the best designs are those in Fig. 14*a*, *c*. If it is necessary to reduce size and weight, the compact designs shown in Fig. 14*l-o* are preferable.

2.5. Method of Inversion

Among the large number of methods devised to facilitate the design there is the method of inversion. Its essence consists in inverting the function, shape and position of parts. In assemblies it is occasionally preferable to invert the functions of parts, e.g., to turn a driving element into a driven one, to turn a leader into a follower, to make a female component out of a male one, to convert a stationary part into a movable element, etc.

Sometimes it becomes advisable to invert shapes of parts, e.g., to change a female taper to a male one, a convex spherical surface to a concave one. In other cases it may be more suitable to shift some elements from one part to another, for instance, to move a key from a shaft to a boss, or a striker from a lever to a pusher.

Each time the design will acquire new qualities. The designer must carefully evaluate all the advantages and drawbacks of the original and inverted alternatives, taking into account durability, size, suitability for industrial production and convenience of operation, and then choose the most suitable version.

The method of inversion is an inalienable instrument of thought for every experienced designer. It significantly lightens the search for solutions as a result of which a rational construction is found.

Some typical examples of inversion in mechanical engineering are shown in Fig. 15.

Rod drive (Fig. 15*a*). In sketch *I* the rod is actuated by a fork lever acting through the agency of the rod pin. In the inverted design *II* the pin is transferred to the lever. This results in a reduced transverse force acting on the rod. However, the design shown in sketch *I* has smaller sizes.

Tappet drive (Fig. 15*b*). In design *I* the striker of the rocker arm is flat and the tappet plate—spherical. In the inverted scheme *II* the rocker arm is spherical and the tappet plate—flat. The result is reduced transverse forces acting upon the tappet. Moreover, the

striker can be made cylindrical, thus assuring a linear contact in the joint, whereas in design *I* the contact is point.

Connecting rod and gudgeon pin (Fig. 15*c*). In scheme *I* the gudgeon pin is secured to the connecting rod and runs in the fork bearings; in the inverted design *II* the pin is fixed to the fork and the bearing is in the connecting rod. In this case the inversion reduces overall dimensions and lightens the work of the bearing acquiring greater rigidity.

Nipple unions. In sketch *I* (Fig. 15*d*) the nipple is tightened by an internal captive nut, but in the inverted version *II* an external nut is used. As a result, axial dimensions are decreased, and the radial ones increased. The alternative shown in sketch *II* is more convenient to tighten.

In the nipple union presented in Fig. 15*e* the female taper on the nipple (version *I*) has been changed to a male taper (version *II*), thus reducing the overall dimensions.

Spherical pipe joint (Fig. 15*f*). The spherical surface of the joint (version *I*) can be substituted by two such surfaces (version *II*); advantages: smaller dimensions and lighter weight. However, the manufacture of such a joint is more complicated, because it is necessary to make the two spherical surfaces truly concentric.

Fixing a turbine blade (Fig. 15*g*). Changing the female fork blade fixing on the T-shaped male tenon of the rotor (scheme *I*) to a female tenon in the rotor and male tenon on the blade (scheme *II*) lessens the weight, gives a greater rigidity and eases the manufacture of the blade base.

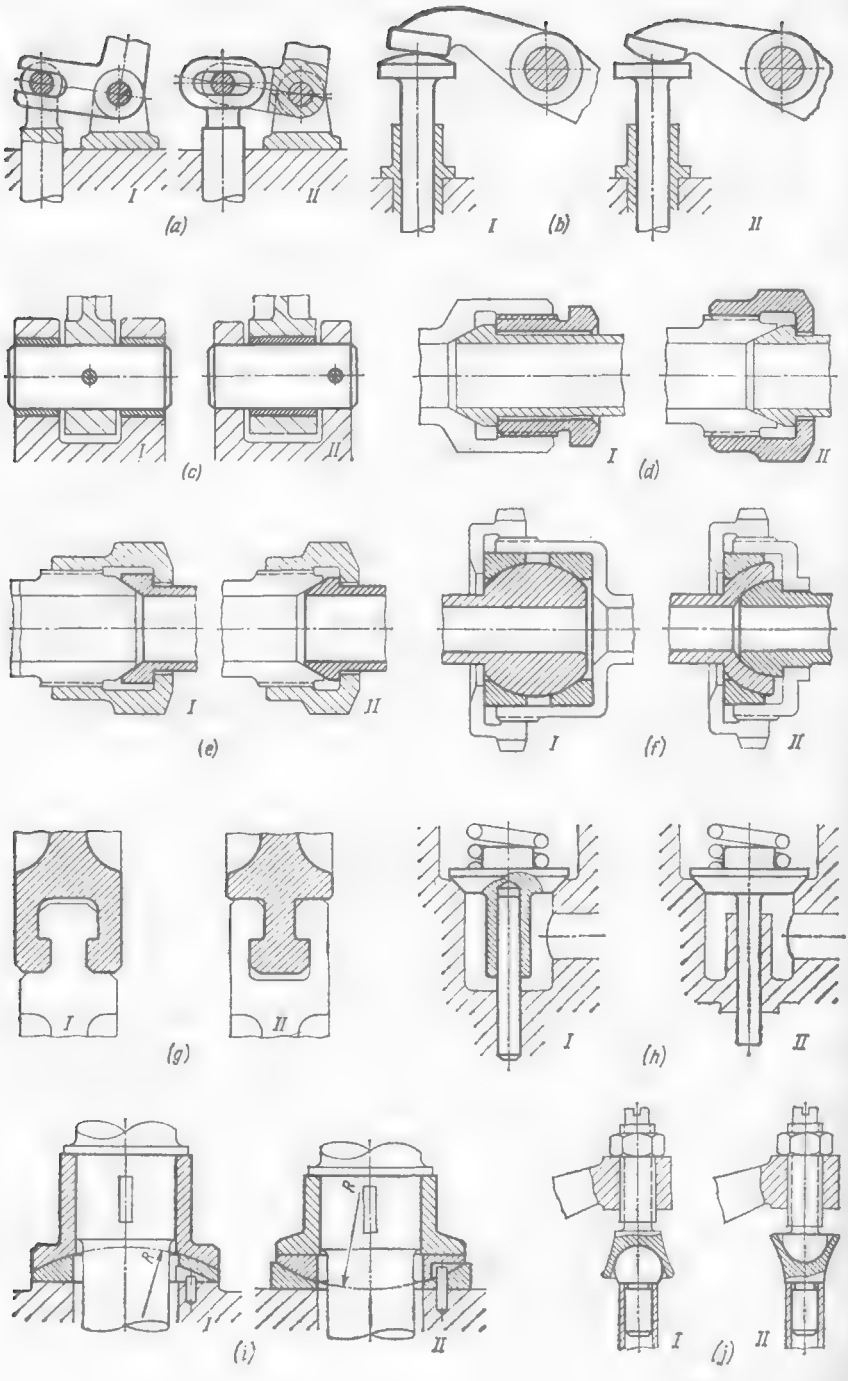
By-pass valve (Fig. 15*h*). Better precision is obtained if the valve is piloted not by a rod press-fitted into the valve body (scheme *I*), but by a shank sliding in the valve body hole (scheme *II*). This is because the pilot hole and valve seat can be machined in one setting and with better concentricity.

Self-aligning thrust bearing (Fig. 15*i*). In the bearing shown in sketch *I* the thrust journal runs on a spherical surface, and in sketch *II*—on flat surface, which is more advisable. The two schemes differ also in the position of self-alignment centres.

Rocker drive (Fig. 15*j*). Transferring the spherical surface from the rod (scheme *I*) to the striker (scheme *II*) improves lubrication, as the oil from the mechanism's interior will collect in the cup-shaped tip of the rod. In version *I* penetration of oil into the joint is almost excluded.

Vee way (Fig. 15*k*). The design given in sketch *II* is more advantageous as far as lubrication is concerned. The groove will retain lubricant, thus adding to the machine's durability and accuracy of movement.

Spring (Fig. 15*l*). In scheme *II* the tension spring is changed to a compression spring with a reverse. The compression spring is



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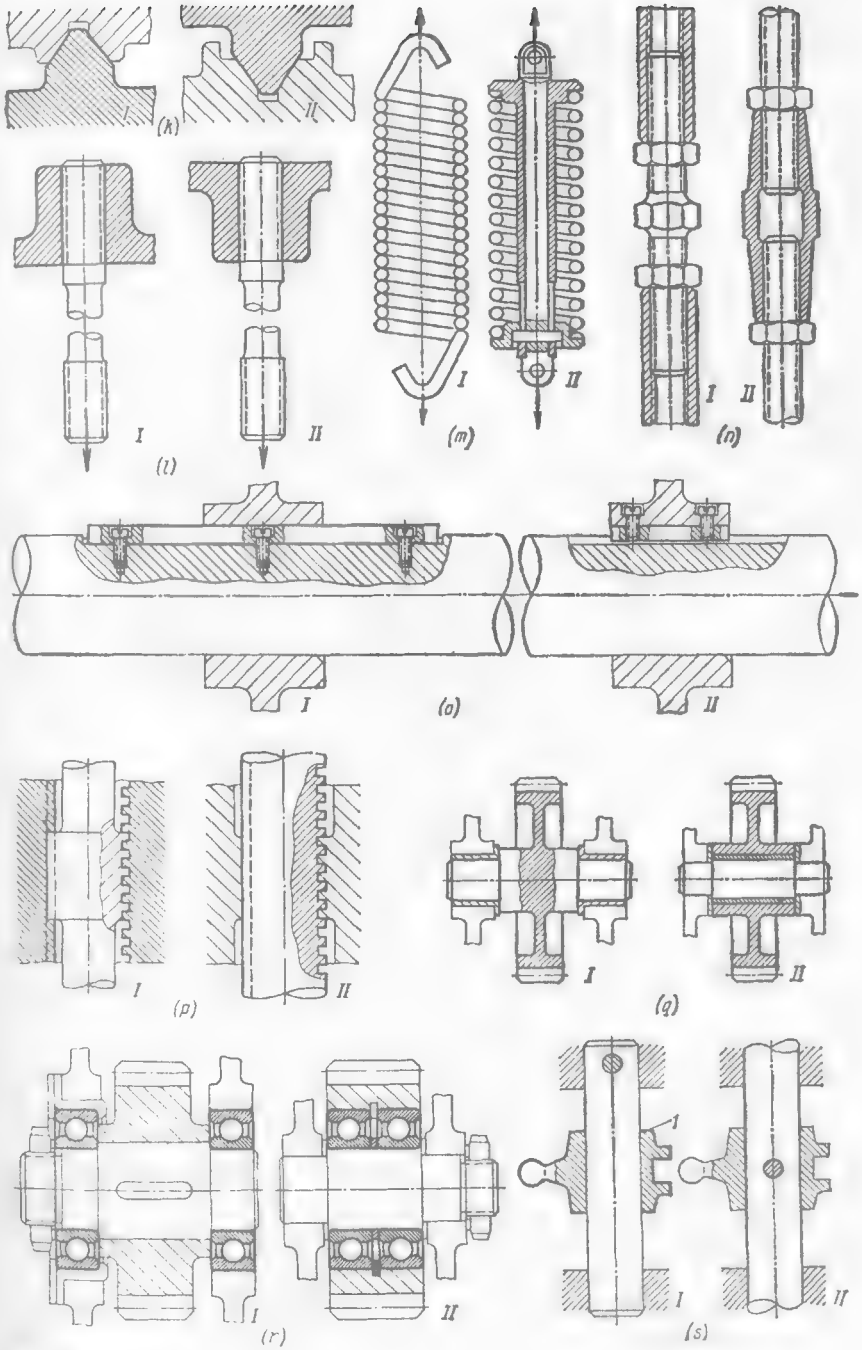


sion of units

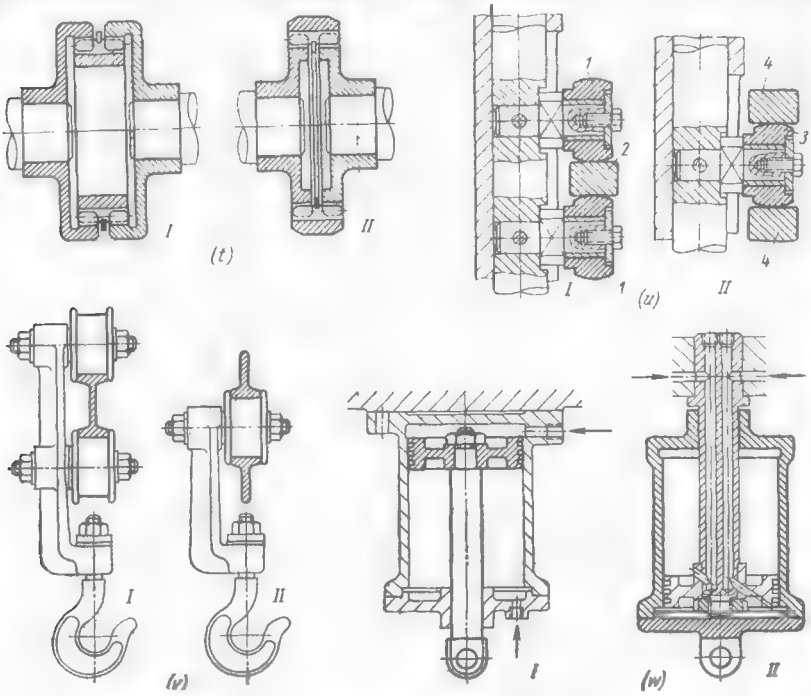


Fig. 15. Inversion of units (cont'd)

stronger and longer lasting than the tension spring in which the tug ends are often stretched. The inverted design is more reliable, but is more complex and heavy than the conventional design (scheme *I*).

Turnbuckle (Fig. 15*m*). This can be made in the form of an externally threaded rod (version *I*) or as a threaded sleeve (version *II*). In the latter case the overall dimensions are reduced by the hexahedron length which in scheme *I* is longer.

Stud securing boss (Fig. 15*n*). Version *II* is more advantageous in terms of the screwed joint strength since the boss of enhanced compliance distributes thread loads more uniformly.

Guide key (Fig. 15*o*). When secured in a hub (scheme *II*), the key will make the design more practical. The difficulty of mounting a long key is avoided since it is easier to make a precision slot in the shaft than fit a long key to the same tolerances.

Lifting screw (Fig. 15*p*). It is more advantageous technically to cut a long thread on a rod (version *II*) than in a nut (version *I*). With the same thread diameter and connection length, the rod shown in the second version will be stronger.

Idle gear (Fig. 15*q*). The arrangement presented in sketch *II* is better as far as dimensions and manufacturing conditions are

concerned. Thus, bearing operating conditions are improved as the supports are more rigid. Furthermore, when the gear is installed according to mode *I*, the shaft, affected by the drive, will be subjected to a cyclic load, while in mode *II* this will be a static load.

Should the gear be mounted in antifriction bearings (Fig. 15*r*) then one must bear in mind that the bearings' durability in scheme *II* (in which the outer race is rotating) will be worse than that of the bearings in scheme *I* (in which the inner race runs).

Slider (Fig. 15*s*). In the first version slider *1* slides along a stationary rod, whereas in the inverted scheme *II* the slider is fixed to the rod which moves in its guideways. This inversion gives improved directional movement.

Splined coupling (Fig. 15*t*). Scheme *II* is better than scheme *I* in terms of axial dimensions and manufacture (internal splines are usually broached).

Actuation of rod by a disk master-form (Fig. 15*u*). In version *I* the rod is actuated by the two rollers *1* mounted on the rod, which are rolling over master-form *2*; in version *II*—by one roller *3* positioned between two master-forms *4*. The latter construction is more compact and eliminates the drive backlash as the clearance between the master-forms and roller can be adjusted.

Suspended hook trolley (Fig. 15*v*). In scheme *I* the trolley has two pairs of rollers (the Figure shows only one pair) rolling on a monorail. In the inverted scheme *II* one pair of rollers travel between two rails. The second scheme is more compact, but requires two rails.

Hydraulic cylinder (Fig. 15*w*). In design *I* the cylinder is stationary, and its piston and drive rod move. In the inverted design *II* the piston and rod remain stationary, and the cylinder fitted with a driving fork moves over them. The advantage of scheme *II*: drive can be effected from any point along the cylinder.

2.6. Composition Methods

Generally composition has two stages: draft and operational. During the draft composition the main arrangement and general design of a given unit are elaborated (sometimes several versions). After discussing the draft the working scheme is composed, defining more accurately the design of the unit and serving as the initial material for furthering the project.

During composition it is important to separate the main items from the secondary and, accordingly, establish the correct order of development and construction.

An attempt to compose all elements of a design at a time is a typical error characteristic of young designers. Having received the assignment presenting fully the purposes and parameters of

the projected design, the designer often begins at once to draw the complete design in all its details, depicts all the design elements to compose a picture as if it were an assembly drawing, a technical or working specification of the project. Such a method makes the design an irrational one, as it forms a mechanical string of constructional elements and units, poorly arranged as is well known.

It is necessary to begin the composition with the solution of the main design problems, i.e., selection of rational kinematic and power schemes and correct sizes and shapes of parts and determination of their most suitable relative positions. Any attempt to fully describe parts at this stage of design is not only useless, but harmful as it draws the attention from the main problems of the composition and confuses the logical course of developing the design.

Another fundamental rule of composition—parallel development of several design alternatives, their analysis and selection of the best version. It is a mistake if the designer at once sets the direction of designing by choosing the first type of design which occurred to him, or a commonplace alternative. The designer must analyse carefully all feasible versions and choose the one most suitable for the given conditions. This requires much work and the problem is not at once solved, and sometimes only after long investigations.

Full development of each alternative is not necessary. Usually hand pencil sketches are sufficient to get an idea of the prospects of the variant and decide on the question of whether it is advisable to continue with this particular alternative.

Composition must always be accompanied with calculations, if only tentative approximations. Main design components should be calculated not only for strength, but also for rigidity.

Never rely on the eye when selecting dimensions and shape of parts. Of course, there are very skillful designers who almost without mistakes can establish sizes and cross-sections assuring stress levels acceptable for the given branch of engineering, but this is a doubtful virtue. Copying trite shapes and keeping to traditional stress levels, one cannot create progressive designs.

To wholly depend upon calculations is wrong. In the first place the existing techniques of strength calculations do not consider many factors influencing the fitness of the design and secondly, there are some parts (e.g., housings) which cannot be calculated at all. Thirdly, other factors besides strength affect the sizes of parts. For example, the design of cast parts in the first place depends on casting technology. For parts being mechanically machined it is necessary to consider their resistance to cutting forces and give them a certain rigidity. Heat treated parts should be large enough to avoid buckling. Dimensions of controls components should be chosen so as to ensure manipulation convenience.

Thus, alongside calculations, the designer must refer to existing designs and introduce, if necessary, well-founded corrections.

Another prerequisite for correct designing is a continuous consideration of the manufacturing problems; at the very beginning the component should be given a technologically rational shape. A skilled designer composing a part at once makes it easy to produce. Young designers should constantly consult production engineers.

Composition must be carried out on the basis of standard dimensions (fitting diameters, sizes of keyed and splined connections, diameters of threads, etc.). This is particularly important when composing units with several concentric fitting surfaces, as well as stepped parts, whose shape to a large extent depends on the gradation of diameters.

At the same time maximum unification of standard elements should be sought for. Chosen elements unavoidable in the design of main parts and units should be utilized in the remaining parts of the design as much as possible.

In the process of composition the designer must take into account all the conditions defining the operability of the machine, develop the systems of lubrication and cooling, assembly and disassembly, and attachment of adjacent parts (drive shafts, pipings, electric wires, etc.); provide for convenient maintenance, inspection and adjustment of the mechanism; choose correct materials for the main components; think out methods for improving the machine's durability and wear-resistance of rubbing surfaces, and methods of corrosion protection; investigate and determine the limits of the machine when operating under forced conditions.

Composition does not always proceed smoothly. Often during designing some small defects, overlooked in the first estimates, are revealed. For their elimination it sometimes turns out necessary to return to schemes rejected earlier or to develop new ones.

Some units are not always successfully designed from the first. This should not discourage the designer—he has to devise some "tentative" alternatives and raise the design to the required level in the process of further activities. In such cases it is useful, as Italians say "*dare al tempo il tempo*" (to give time to time), i.e., to take a breathing space, after which, as a result of subconscious thought, the problem is often solved. After a while the designer looks at the outline drawing in another light, and sees the mistakes made at the first stage of the development of the main design idea.

Sometimes the designer unintentionally loses his objectiveness and does not see the drawbacks of his favourite variant or the potentialities of other versions. In such cases impartial opinions of outsiders, the advice of elders and co-workers, and even their fault-finding criticism turn out to be most opportune. Moreover, the sharper the criticism, the greater is the benefit derived by the designer.

At all stages of the composition manufacturers and operators should be consulted.

A general rule is such: the wider the discussion of the composition and the more attention the designer pays to useful advice, the better becomes the composition and the more perfect the design.

Do not spare time or effort on working up the design. The cost of designing is only a small portion of the machine manufacturing expenditures (excluding machines produced individually or on a small-batch production basis). In the final analysis, the greater the development work on the design, the more are the savings in the machine cost, time of manufacture and finishing, the better its quality and the greater the economic gains over the machine's service life.

2.7. Composition Procedures

If the overall dimensions of a project allow, its composition is best of all drawn to a 1 : 1 scale. This facilitates selection of required dimensions and sections and enables a realistic presentation of machine part proportions, their strength and rigidity, to be individually and as a whole obtained. In addition such a scale obviates the necessity for a large number of dimensional specifications and simplifies subsequent stages of design, in particular detail drawings, since such dimensions can be taken directly from the drawing.

Tracing to a reduced scale, particularly at reductions exceeding 1 : 2, strongly obstructs the composition process, distorts proportions and clarity of the drawing representation.

If the dimensions of design deny the use of a 1 : 1 scale, then at least individual units and groups should be composed to natural size.

Composition of simple components may be developed in one projection if the drawing explains it sufficiently well and clearly. The form of the construction in a cross-sectional plane will be filled by a three-dimensional imagination. However, when applied to more sophisticated compositions, this method may cause serious errors; therefore, the design must be developed in all necessary projections and cross sections.

The fulfilment of an arrangement drawing is, a continuous process of research, trial, approximation, development of alternatives, their comparison and rejection of unsuitable versions. Sketches should be lightly traced with a pencil during the composition corrections, follow one another, which means that a rubber is applied more often than a pencil.

Cross sections can be left unhatched, but if hatched, then only by hand. Do not waste time on drawing parts. Typical components and units (fasteners, packing, springs, antifriction bearings, etc.) should be simply depicted, examples of this are shown in Fig. 16.

Contour outlining, hatching, provision of a legend and particulars of small parts are made at the final composition stage, when the arrangement is ready for discussion.

There is a school for making hand-drawn compositions, the design is drawn with a pencil on a sheet of squared paper. The author always used this method, and considered that such composition had great advantages as to capacity, flexibility and easiness of

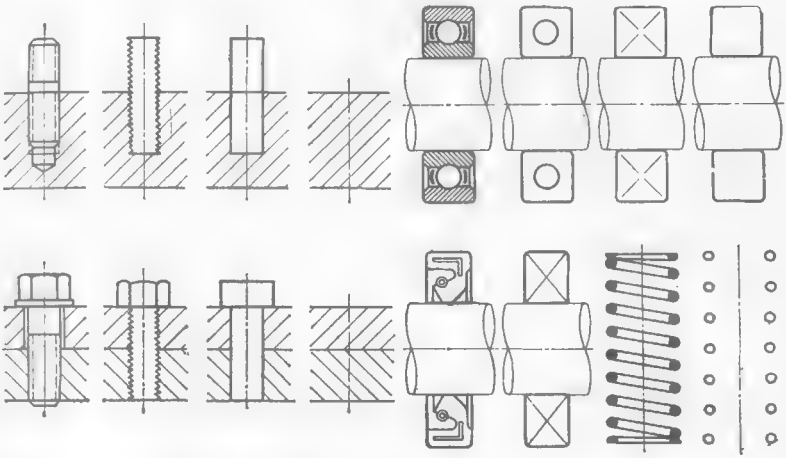


Fig. 16. Simpilified sketches of common parts as used on general arrangement drawings

introducing corrections. The method almost practically excludes any errors in connecting dimensions and ensures comprehension of the part sizes.

This method is especially useful for showing smooth outlines characteristic of modern designs.

The method is convenient for designers having certain aptitudes for drawing. Some designers are capable, when applying this method, of preparing in a few hours complete arrangements, which can be handed over for detailing.

2.8. Design Example

To illustrate the technique of composition let us consider the design of a centrifugal water pump. This device has certain specific features, which influence the techniques and sequence of composition. In the case under consideration there will be rather a stable original basis in the form of a sketch, prepared by the draft design department, describing the hydraulic parts of the pump. It remains

for the designer to embody the draft design in metal. Very often the designer is given only a most general scheme of the machine without basic sizes. Sometimes the designer begins his work with knowledge of only the purpose and specifications of the machine without even a scheme of its future construction. Under such circumstances he has to begin with thinking over the main concept of the design and investigate versions of design schemes. Only after this does the composition take shape.

The technique of composition described below is not the only one possible. Like any other creative process the process of arrangement composition is subject to and in many instances is largely dependent on the experience, skill and aptitudes of the designer. There may be different ways in composition, sequence of developing the design and also solutions to the problems occurring during design. The technique, illustrated below, is for example only and is intended to illustrate the basic regularities to this or that degree inherent in any composition process. These are:

sequence of development, revelation at the first design stage of the basic constructional elements ignoring details of construction;

consideration of several alternatives, choosing the best on the basis of comparison of technology and operational practice;

parallel with designing, tentative calculations for strength, rigidity and durability;

building-in at the first composition stages machine reserves for future development and the limits of its operation in a forced regime;

making the design suitable for industrial production, implementing successively unification and standardization;

developing practicable assembly and disassembly schemes;

careful examination of machine parts and units for durability and operational reliability.

In the next example the results of each composition stage are given as separate drawings. A young designer may have the impression that the composition process consists of the successive preparation of such drawings. In actual fact, it is one and the very same composition drawing which is continuously being amplified and clarified as the development proceeds until it gains finality.

In the interests of better representation in the drawings below small details are presented (not always) in full. In reality in the composition process they are represented simply by conventional symbols and sometimes not at all.

In the book a full account of the variants has to be drawn so that detailed explanation may be given when comparing the advantages and disadvantages of the different constructional solutions. During design the greater part of the variants are thoughtfully compared by the designer who at once rejects unsuitable solutions

and only sometimes sketches by hand other variants without even observing scales.

Sometimes the designer cannot explain why he chose or rejected one of the design solutions and only limits his remarks by laconically saying: "I like (or dislike) it". Thus what appears to be tasteful justification to an experienced designer, in fact, conceals an intuitively correct appraisal of the instructional, operational and other complications, which he carries out when rejecting certain lines of thought.

However, the process of comparison and selection of alternate designs progresses much more rapidly than it may seem from the descriptions and illustrations given below.

Much time has to be spent on the solution of complex or new design problems which may occur in a designer's work requiring creative work searches for similar supporting examples from various engineering branches and sometimes the organization of experiments, which, depending on the allotted design time, are carried out quickly or to the dictated circumstances.

(*a*) *Initial Data*

The starting material for designing the pump is a sketch of the pump hydraulic cavity, with basic dimensions (Fig. 17). This must

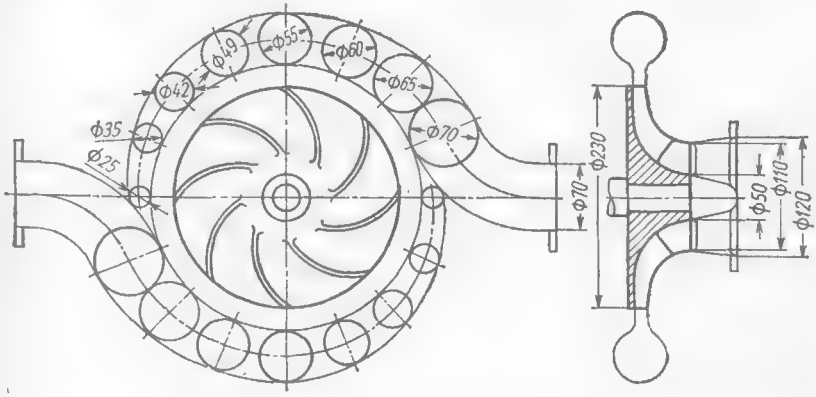


Fig. 17. Sketch of pump hydraulic cavity

be a single-stage pump with an axial inlet and cantilevered impeller. In the intake pipe is fitted a guiding device assuring axial flow of the water stream to the impeller.

The pump is driven by an induction motor ($n = 2950$ rpm). The impeller peripheral speed is 35.5 m/s, the design head, 50 m

WG, and the pump delivery, 40 l/s. The pump has two symmetrically arranged discharge pipes, each 40 cm² in cross-sectional area.

The calculations supplied with the sketch give the number and arrangement of vanes (eight vanes curved to suit direction of impeller rotation), the profile of the impeller flow portion, cross sections of discharge volutes at circumferential angular positions. The pump service life is preset 10 years of operation to a double-shift schedule).

The calculated pump durability is equal to the product of the pump service life in hours and the shift and off-days factors (no factor for repair downtimes is introduced as a repairless operation is assumed). Hence,

$$h = \eta_{shift} \cdot \eta_{off\text{-}d} \cdot H$$

where $H = 10 \cdot 365 \cdot 24 = 87\,600$ h is the nominal service life;
η_{shift} = shift factor (≈ 0.6 for double-shift operation);
$\eta_{off\text{-}d} = 0.8$ is the off-days factor

The calculated durability

$$h = 0.6 \cdot 0.8 \cdot 87\,600 \approx 40\,000 \text{ h}$$

(*b*) *Shaft Supports*

It is advisable to begin with selection of the type, dimensions and arrangement of impeller shaft supports. The latter are assumed to be ball bearings which advantageously differ from plain bearings, as the lubrication of the former is much simpler.

The radial load upon the bearings is established from the weights of the impeller and shaft and the centrifugal load due to incomplete static balance of the impeller. In addition, the supports withstand the axial pressure from the operating fluid on the impeller.

On the basis of the preliminary rough estimates assume that impeller weight $G_{imp} = 4$ kgf (with a margin), the weight of shaft and connected parts (bearing inner races, drive flange, clamping nuts) $G_{shaft} = 2$ kgf.

The unbalanced centrifugal force of the impeller may be found from the magnitude of the static unbalance. Assume that the value of the static unbalance at the impeller periphery is 5 kgf. Then, the maximum possible unbalanced centrifugal force will be

$$P_{ctfg} = \frac{0.005\omega^2 R}{9.81} = \frac{0.005 \cdot 310^2 \cdot 0.115}{9.81} = 5.5 \text{ kgf}$$

The maximum radial force acting upon the impeller in the plane of its centre of gravity will be

$$P = G_{imp} + P_{ctfg} = 4 + 5.5 = 9.5 \text{ kgf}$$

The load R_1 on the bearing nearest to the impeller is

$$R_1 = P\left(1 + \frac{l}{L}\right) \tag{2.1}$$

where l = distance from the impeller centre of gravity to the front support;

L = distance between the supports

The load R_2 on the second bearing will be

$$R_2 = R_1 - P = P\frac{l}{L} \tag{2.2}$$

As is well known the best ratio L/l lies within the range of 1.5-2 (see Fig. 125). Below these values the forces R_1 and R_2 sharply increase; increasing the L/l ratio in excess of 2 does not significantly reduce these forces, but only leads to increases in the axial dimensions of the unit.

Assume $L/l = 1.5$.

Then, according to Eqs. (2.1) and (2.2)

$$R_1 = 1.66P = 1.66 \cdot 9.5 = 16 \text{ kgf}$$

$$R_2 = 0.66P = 0.66 \cdot 9.5 = 6.3 \text{ kgf}$$

To these values it is necessary to add the shaft weight $G_{shaft} = 2$ kgf which is distributed almost equally between the two bearings. Thus,

$$R_1 = 16 + 1 = 17 \text{ kgf}$$

$$R_2 = 6.3 + 1 = 7.3 \text{ kgf}$$

For the sake of unification assume the two bearings to be of the same type. As the rear bearing bears a lighter load, it is better to make this bearing fixed, so that it may take up axial forces.

(c) *Counterbalancing of Impeller Axial Force*

The rear faces of open impellers receive the full force of the hydrostatic pressure, being built up at the discharge outlet (in our case $p = 5$ kgf/cm^2). The force, acting in the opposite direction, is substantially less, because the pressure upon the impeller disk from the vane side changes to a square law, beginning from vacuum in the intake pipe up to 5 kgf/cm^2 at the impeller discharge point. As a result, an axial suction force, directed to suction, occurs, which in the considered case reaches approximately 1000 kgf. This force can be eliminated by employing an enclosed double-disk impeller with a bilateral seal, and by introducing bleed holes in-between suction and pressure cavities (Fig. 18). Such a system fully counterbalances the hydrostatic pressure upon the impeller as it is subjected to identical pressures (5 kgf/cm^2) on both sides.

Apart from hydrostatic forces the impeller is affected also by the reactive force of the jet, which, at the intake, is directed against the suction. However, this force is not large and can therefore be neglected.

The condition for hydrostatic balance requires that both seal diameters be the same and the total area of bleed holes be at least equal to the circular clearance in the seal.

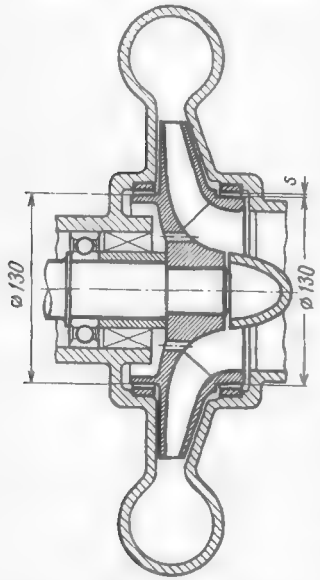


Fig. 18. Impeller with counterbalanced axial force of hydrostatic pressure

Assuming that seal diameters $D_{seal} = 130$ mm, radial clearance $s = 0.1$ mm, number of the bleed holes $n = 8$ (equal to the number of the vanes), we obtain

$$n\frac{\pi d^2}{4} \geqslant 0.1\pi D_{seal}$$

hence

$$d \geqslant \sqrt{0.05 D_{seal}} \geqslant 2.5 \text{ mm}$$

With some margin we take the value of d at 5 mm.

The seals are made in the form of cylindrical bosses on the impeller disks and fit, with clearance, into rings press-fitted into the pump casing. To allow for possible touching of sealing surfaces the rings are made of an antifriction material (soft bronze, grade Бр. ОЦС).

The seals can also be made of fluorocarbon or silicone plastics, which do not swell in water. However, the high cost of these materials should be kept in mind and their high coefficient of linear expansion makes the problem of ring fixture in the casing much more complex.

(*d*) *Durability of Supports*

Tentatively assume that the impeller shaft diameter $d = 40$ mm and that the supports are, in effect, single-row ball bearings (light series) whose work capacity coefficient $C = 39\,000$.

The work capacity coefficient providing for the required pump durability

$$C = Rk_\sigma (nh)^{0.3}$$

where R = load on bearing (in this case the greatest bearing load $R = 17$ kgf)
k_σ = bearing duty factor (assume $k_\sigma = 1.5$)
n = shaft speed, rpm ($n = 2950$ rpm)
h = given durability ($h = 40\,000$ h)

Consequently, $C = 17 \cdot 1.5\ (2950 \cdot 40\ 000)^{0.3} = 6800$. Thus, the selected bearings will satisfy (with an ample safety factor) the pump durability period and enable larger loads and increased speeds to be used in the event of forced working.

To consider bearings for durability alone is not sufficient, as the calculations assumed normal working conditions for the bearings. Mistakes in assembly, too little or too much lubrication may bring all calculations to nought and cause premature wear or even failure long before their service life has expired.

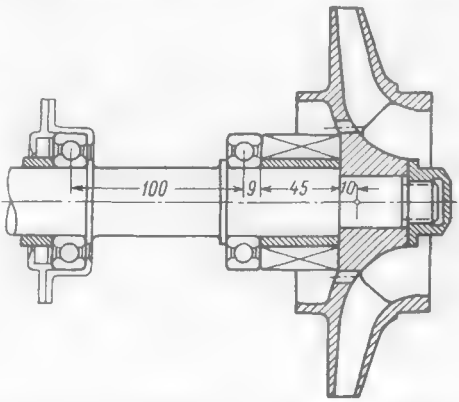


Fig. 19. Arrangement of impeller shaft supports

(e) *Spacing of Supports*

Whith the chosen ratio $L/l = 1.5$ the spacing between the supports will entirely depend on the overhang value l of the centre of gravity of the impeller relative to the front support. The latter value is determined by the arrangement of the seal between the front bearing and the pump hydraulic cavity.

On the basis of the preliminary estimates assume that the seal length is 45 mm and the distance between the front seal face and the impeller centre of gravity, 10 mm. The bearing width is 18 mm. Hence, the total length of the overhang

$$l = 45 + 10 + 9 = 64 \text{ mm}$$

and the distance between the supports

$$L = 1.51 \cdot l \approx 100 \text{ mm}$$

This stage of the pump design ends with the drawing of a sketch of the impeller shaft, which also illustrates the support spacing (Fig. 19).

(f) *Discharge Volutes*

The cross sections of volutes can be positioned so that the extreme internal points of the sections are equispaced about the impeller circumference.

Then the centres of the cross-sections are arranged along a spiral described by the equation

$$\rho = \frac{D_{imp}}{2} + s + \frac{d_0}{2}\sqrt{\frac{\varphi 0^\circ}{180^\circ}}$$

and the extreme external points of the cross-sections will follow the spiral described by the equation

$$\rho' = \frac{D_{imp}}{2} + s + d_0 \sqrt{\frac{\varphi}{180°}}$$

where D_{imp} = diameter of impeller;
s = distance at which the internal points of the cross-sections are located from the impeller circumference;
d_0 = diameter of the volute discharge cross-section;
ρ, ρ' and φ = current coordinates

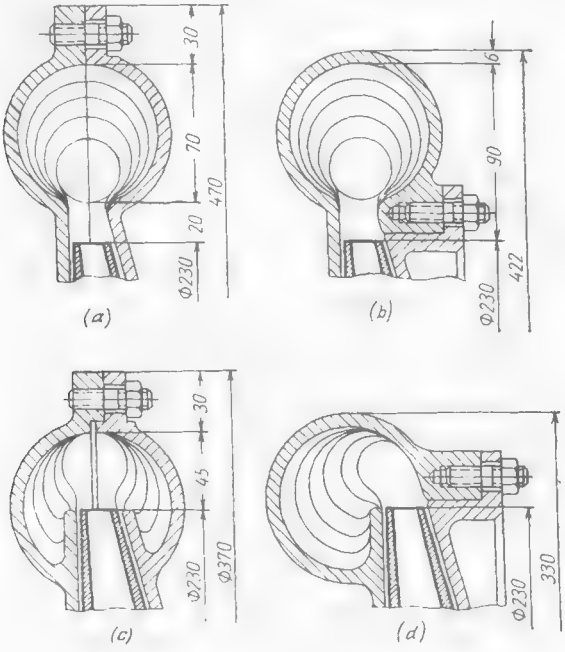


Fig. 20. Design variants of pump discharge volutes

Such a spiral volute with a split symmetrically along the cross-section (Fig. 20*a*) ensures simple coreless moulding and enables easy cleaning of the volute cavity.

Disadvantages:

the split plane intersects the volute discharge pipes; as a result, a T-joint very difficult to seal is formed in the discharge pipe flanges and in outlet pipe joints associated;

the volute halves can be fixed relative to each other by fitted dowel pins only; centring with the aid of a cylindrical spigot is excluded;

the volute radial dimensions are too large (when $s = 20$ mm, the maximum size, without discharge pipes, will be 470 mm).

In the design shown in Fig. 20*b* the volute is a complete casting. The impeller is assembled through detachable cover. The discharge pipes are one piece. The cover is centred relative to the casing by a locating spigot diameter. The volute size is somewhat reduced due to elimination of the peripheral flange (maximum size is now 422 mm). The impeller hydraulic cavity is enclosed and shaped, during pouring, by moulding cores. The cavity can be cleaned only by hydropolishing (water jet containing suspended abrasive particles).

In the design in Fig. 20*c* the external points of cross-sections are spaced over the circumference whose radius is equal to the minimum initial radius of the volute. Outwardly the centres of cross-sections are gradually shifted towards the pump axis, following a spiral pattern described by the equation

$$\rho = \frac{1}{2}\left(D_0 - d_0\sqrt{\frac{\varphi}{180^\circ}}\right)$$

where D_0 = volute outer diameter;

d_0 = diameter of the volute discharge cross-section

In the last portions of the volute the impeller, together with the walls binding it to the casing, penetrate into the volute section. The overall size of volute is sharply decreased (370 mm). The joint line is along the volute cross-section plane of symmetry. The volute halves are positioned by a locating spigot diameter (interrupted in the discharge pipe areas). The discharge pipes are intersected by the split plane.

The design disadvantage is that the water, as it flows to the output from the impeller, bifurcates forming in the exit part of the volute two spiral vortices causing greater hydraulic losses.

The discharge pipes can be made in one piece if the volute section is offset from the axis of impeller symmetry (Fig. 20*d*). In this case the impeller mounting is accomplished through a cover. The elimination of the peripheral flange decreases the volute size still more (the maximum size is 330 mm).

Offsetting the volute sections will cause a water vortex but hydraulic losses will be less here than in the design displayed in Fig. 20*c*.

Finally, comparison of the schemes preliminarily establishes that the best design is given in Fig. 20*b*, as it possesses significantly fewer drawbacks and a comparatively larger number of advantages.

(g) *Hydraulic Cavity*

An arrangement drawing of the pump hydraulic cavity consisting of volutes, cover and intake pipe guiding device, is presented in Fig. 21. The guiding device consists of radial vanes cast on to the pipe walls and to a streamlined shaped central boss which assures a smooth water approach to the impeller.

The cover-to-volute joint is lined with rubber cord *a* placed into a ring-shaped recess cut in the centring spigot. Cover dismantling is simplified by recesses *b*, provided for insertion of dismantling instruments, which are disposed in-between stud bosses of the casing.

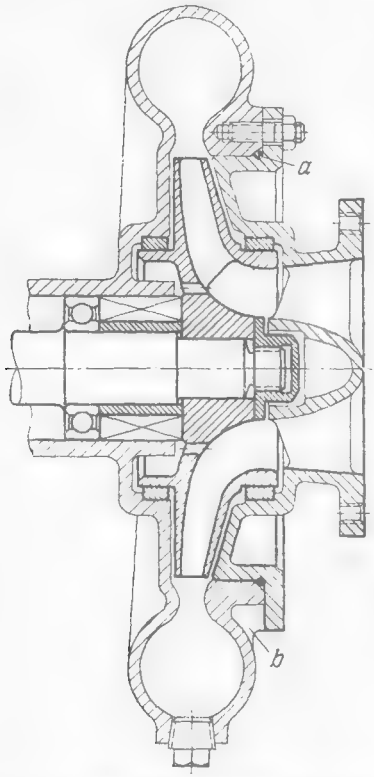


Fig. 21. General arrangement drawing of pump hydraulic cavity

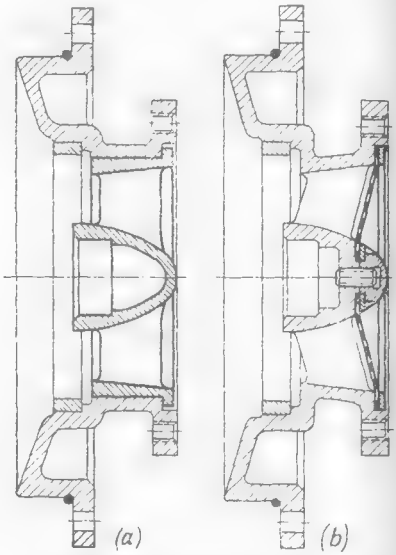


Fig. 22. Guide device variants

During development of the prototype pumps with detachable guiding device can be made to help manufacture (Fig. 22*a*), but for mass production the simpler version, shown in Fig. 22*b*, is preferred.

A filter should be provided when the pump is expected to work in contaminated waters (Fig. 22*b*). A tapered threaded drainage plug should be located below the volute in the longitudinal plane of volute symmetry (Fig. 21).

Water drainage can be automated by closing the drain hole with a spring-loaded valve. As the pump is started, the valve is closed by the pressure of water in the volute; as soon as the pump is stopped, water pressure drops and the spring again opens the valve, thus connecting the volute directly to drainage piping. The possibility of introducing such a device should be noted and drawn (Fig. 23) and later discussed at the final development stage.

The remaining pump elements (design of discharge pipes, lugs separating discharge pipes from volutes, etc.) which seem rather simple will be considered at the stage of their design.

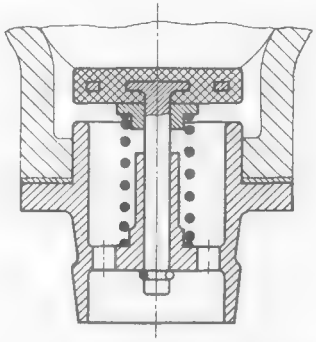


Fig. 23. Automatic drainage device

(*h*) *Hydraulic Cavity Sealing*

The seal separating the hydraulic cavity from the bearing is a very important unit on which to a significant degree depends the operational reliability and durability of the pump.

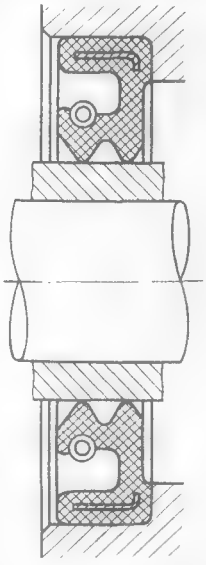


Fig. 24. Sevanite seal

To fully exclude ingress of water from the hydraulic cavity into the oil space, it is better to seal in two stages, positioning one seal on the water side and one on the oil side with an open space between, having bleed holes open to atmosphere.

The water stage, for the most effective sealing, must be packed with a face seal since it possesses the property of self-running-in in contrast to conventional packing glands which require periodic retightening. The oil side should be packed with a sevanite cup seal held in position by a bracelet spring (Fig. 24).

In the first sketch (Fig. 25*a*) the face seal is disk *1* carrying sevanite seal *8*. The disk face is, in effect, the sealing surface. The seal movable part comprises washer *2* rotated by a toothed rim

cut on the inside of the impeller relief packing ring. The washer is constantly pressed to the non-rotating disk by the action of a spring resting against the impeller face. The secondary seal, which prevents water from leaking along the impeller shaft space bush *3*, is made

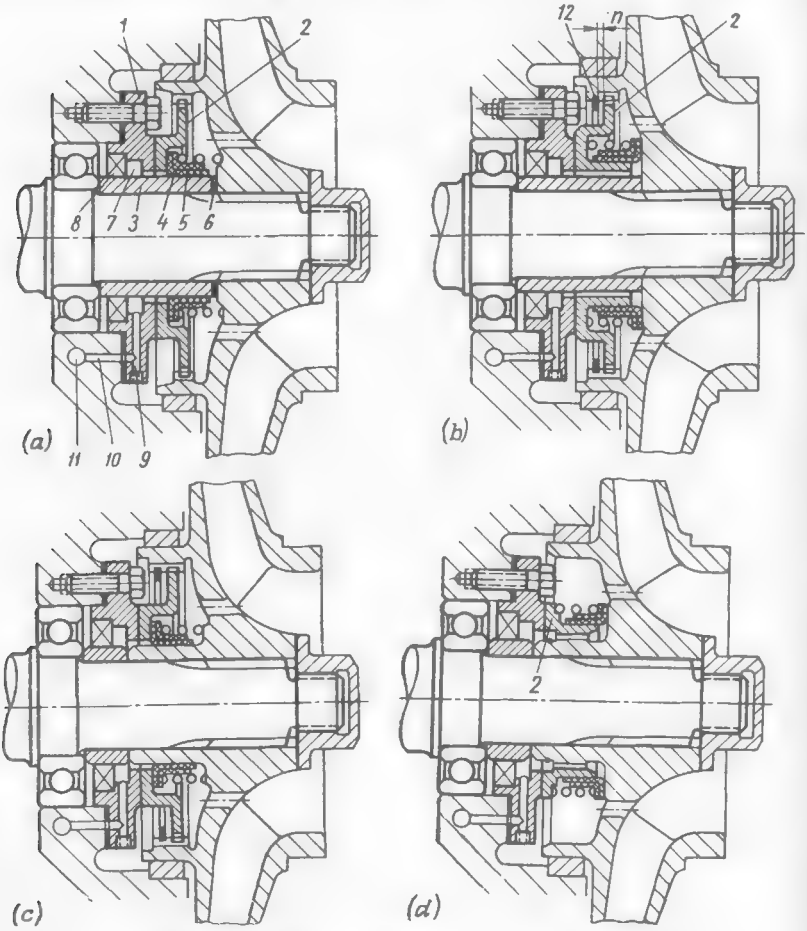


Fig. 25. Design versions of end face seals

in the form of rubber cup *4* tightly encircling the surface of the distance piece. The cup collar is pressed to washer *2* by the same spring acting through steel sleeve *5*. Infiltration of water through the joint between the impeller and distance piece is precluded by ring gasket *6* mounted at the joint.

The intermediate space is formed by cavity *7* between the se vanite seal *8* and wall of disk *1* and is connected by radial hole *9* in the disk

flange with the radially drilled hole *10* in the casing which, in its turn, communicates with atmosphere through hole *11*. For convenience of inspections, viewing condition of packing (water leakage), hole *11* is extended sideways by a pipe fitted in the casing wall.

Disadvantage of the design: when dismantling the impeller the spring forces sealing washer *2* out of engagement with the impeller and pushes rubber cup *4*, consequently the seal disintegrates. Reassembly of the impeller and seal is rendered difficult for the same reasons.

In the design shown in Fig. 25*b* sealing washer *2* is fixed axially in the impeller with the aid of split spring ring *12* mounted on the toothed rim of the impeller with clearance *n* assuring axial displacement of washer *2*, thus compensating wear of sealing surfaces. The secondary cup seal is fixed on the cylindrical extension of washer *2*. During dismantling the movable sealing unit slips off, as a whole, together with the impeller. Assembly is also easier since the above units can be freely slipped onto the shaft.

In another design (Fig. 25*c*) the secondary cup seal is installed on the cylindrical extension of the impeller hub. The design is better in that the impeller centring on the shaft is more precise.

In both of the cases (Fig. 25*b* and *c*) it is not necessary to provide an additional sealing gasket between the impeller hub and distance piece (Fig. 25*a*, gasket *6*).

The final alternative is shown in Fig. 25*d*. Here in the rotating seal washer *2* is made to move by splines cut on the impeller hub, making also a more compact design.

The possibility of water penetration along the splines is averted by tightening the impeller on the shaft by a captive nut with a sealing gasket between the nut and the impeller hub end faces.

With a unit pressure of 2 kgf/mm^2 on the seal working surfaces the axial force developed by the spring is negligible and can be disregarded when calculating acceptable loads on the fixed bearing.

(*i*) *Mounting of Bearings and Impeller on the Shaft*

The first design sketch of the shaft with bearings, impeller and drive flange assembled is shown in Fig. 26.

The main requirement for reliable mounting of bearings on a shaft is their adequate tightening in the axial direction. Preliminarily we apply the following procedure for fastening the bearings on the shaft: the front (right-hand) bearing is tightened by the captive nut which secures the impeller, pressing the bearing against the shaft shoulder through a distance piece. The rear bearing is tightened through the drive flange hub by the nut which secures the flange.

The drive flange hub shoud be long enough to accommodate the external shaft seal. For the sake of unification we also use a sevanite seal here similar to that in the shaft face assembly. The length of the drive flange hub should also suit dismantling tool lugs to be put behind the flange. An acceptable hub length in the first instance appears to be 25 mm.

The impeller and drive flange are spline-mounted. For the sake of unification the splines of the impeller and drive flange, as well

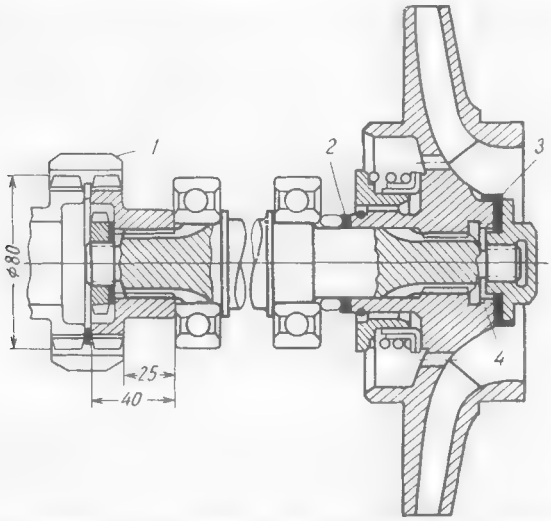


Fig. 26. Shaft, impeller and bearings assembled (general arrangement drawing)

as the threads of the fastening nuts, are similar. The splined connection is assembled by a wringing fit and is centred on the external spline diameter and the side faces of the splines $\left(\text{fit } \frac{A}{W}\frac{U_3}{S_2W}\right)$.

The torque from the electric motor shaft is transmitted to the drive flange by involute splines cut in the flange periphery (spline ring width 15 mm, diameter 80 mm).

A similar flange is mounted on the electric motor drive shaft. The flanges are then joined by splined sleeve *1* which is loosely fit on the splines of the two flanges and held in place axially with a split ring. This coupling transmits high torques with small axial dimensions and enables shaft misalignments to be compensated.

We now introduce improvements: an internal thread *4* is cut in the impeller hub to receive an extractor and washer *2* is installed between the impeller hub and the distance piece for axial adjustment of the impeller inside the casing.

The captive nut is locked by tab washer *3*, whose tabs on one side are bent into slots in the impeller hub and on the other side into slots cut in the edge of the captive nut. The tab washer is made of annealled stainless steel, grade 1X18H9, which also allows the washer to be used as a sealing gasket, preventing infiltration of water into the splines, as well as the nut and extractor threads.

(*j*) *Assembly and Disassembly*

The order of assembly and disassembly is closely linked with the system of installing the bearings on the shaft and in the casing. In principle, two assembly-dismantling techniques are feasible

When applying the first technique, the bearings are installed in the casing by an interference fit, and on the shaft, by a centring or wringing fit. The dismantling procedure will be as follows. First, remove the drive flange, then, moving the shaft to the right, extract it together with the impeller out of the bearings bores (Fig. 27*a*).

Another order of dismantling is possible: first remove the impeller and then, extract the shaft out of the bearings by moving it to the left by the drive flange (Fig. 27*b*).

The above-described procedure avoids the tightening of bearings against the shaft shoulders but requires installation of a distance piece *1* between the bearings. Under such circumstances the impeller must be fixed axially on the shaft by pushing it against the spline step *2*. Both bearings are tightened against the impeller end face by means of the drive flange fastening nut. The tightening force in this case is transmitted to the first (right-hand) bearing through the distance piece.

Disadvantages of the scheme are the following:

after the shaft extraction, the distance piece remains in the pump casing; this makes assembly of the shaft more difficult;

the shaft fitting surface under one of the bearings, as the shaft is being extracted, may be damaged by the inner race of the other bearing.

The worst feature of the arrangement is that the inner races are not interference fitted to the shaft. During long-time operation the fitting surfaces may become worn under the action of radial forces. In principle it is more advisable for slide fits to be used on the outer race diameter where the unit pressures are much less (in this case half).

In the second assembly scheme (Fig. 27*c*) the bearings are fitted on the shaft with interference and are extracted together with the shaft during dismantling. In this event bearings can be tightened against shoulders machined on the shaft. The bearings are individually secured: the front bearing is tightened through the distance

piece by the impeller captive nut; the rear bearing is held in place by the drive flange locking nut.

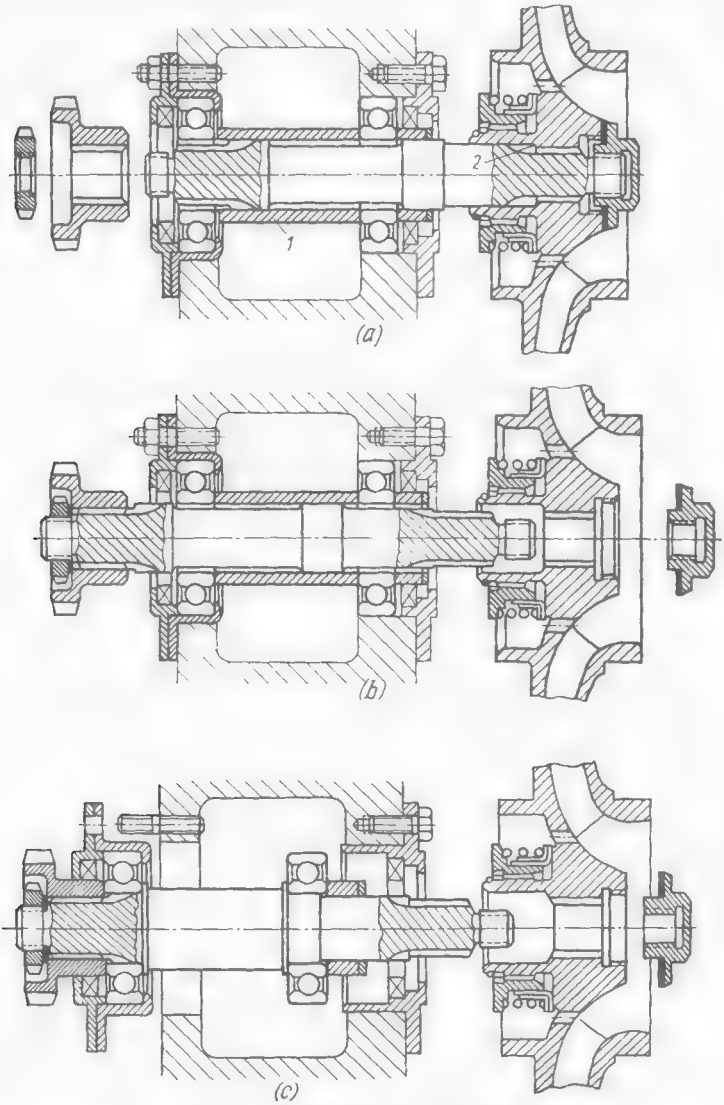


Fig. 27. Dismantling schemes

(*a*) and (*b*) interference fit of bearings in casing; (c) interference fit of bearings on shaft

It is better to mount the bearings inside the casing in reducing sleeves; in doing so, the rear, fixed bearing should be installed in the sleeve with an interference, but the sleeve in the pump casing,

by a slide fit. The front bearing also should be installed in a reducing sleeve by a slide fit. The sleeve, integral with the front seal housing, is installed inside the pump casing by a wringing fit and bolted.

The dismantling procedure is as follows. Detach the impeller from the shaft, undo the bolts fastening the rear seal housing and with the movement to the left extract the shaft together with bearings. The rear bearing is removed out of the casing together with its sleeve and sevanite seal housing. The front bearing seal remains in the pump casing. As the shaft is being extracted, the front bearing freely passes through the enlarged fitting hole of the rear bearing.

When completely dismantling, the bearings are pressed off the shaft, this being easier than extracting bearings from the casing (as in the first scheme).

Comparative analysis of the two schemes shows that the second one is preferable. Therefore the second scheme is taken as the basic one.

(*k*) *Lubrication System*

Generally, the pump bearings run under light loads and comparatively high speeds. The walls of oil retaining housings are very well cooled owing to the proximity of the water flow inside the hydraulic cavity. Under such circumstances splash lubrication is reasonable, employing a low viscosity oil with a gently sloping viscosity-temperature characteristic. A suitable grade is industrial oil grade 12, 12 cSt at 50°C.

When composing the lubricating system the following problems have to be solved:

prevent bubbling and frothing of oil which causes excessive heating and accelerates thermal degeneration;

provide ample oil reserves for prolonged running;

assure a moderate and regular supply of oil to bearings;

avoid excessive bearing lubrication and prevent balls and cages from oil splashes;

provide ventilation of the oil cavity to avoid cavity pressure rise and forcing of oil through seals during warming (starting) periods and formation of vacuum when cooling (stoppage periods);

provide oil drainage and filling facilities;

provide a convenient means for controlling oil level.

The first two of the above-listed problems may in the main be solved by providing a large oil sump in the lower part of the pump casing (Fig. 28). If the sump capacity is insufficient and its size cannot be increased in the axial direction then it may be increased by expanding it laterally.

Protection of bearings from too much oil is achieved by special anti-splash disks *2* installed inside the oil cavity at the bearing side faces.

In the design under consideration a regular supply of oil to bearings is somewhat difficult to maintain. Often an oil bath system filled

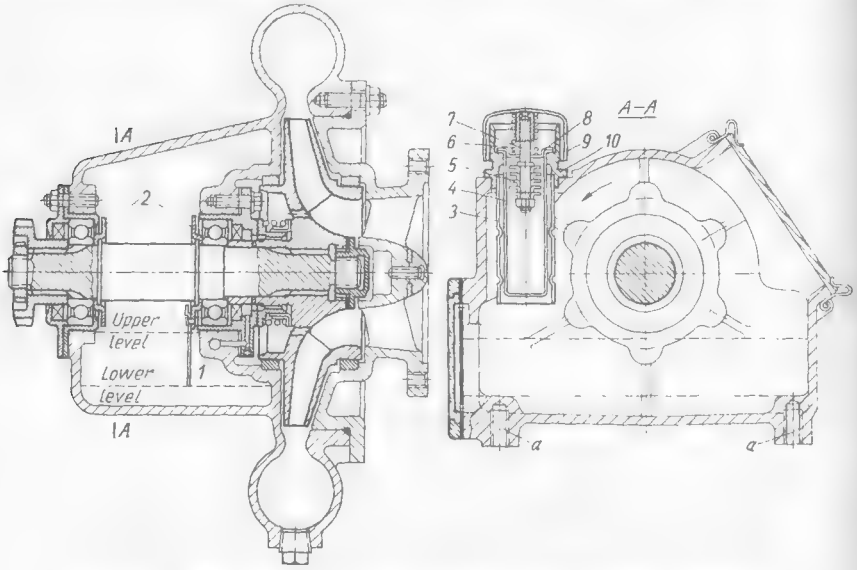


Fig. 28. Pump with oil cavity (general arrangement drawing)

to the level of the lower balls does not solve the problem, as the oil level lowers (due to evaporation of volatiles) and the bearings lack lubrication long before the oil becomes fully exhausted compelling also frequent addition of oil.

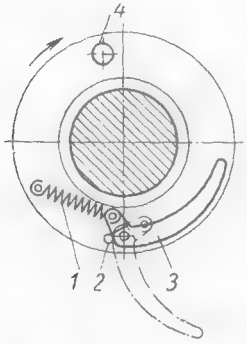


Fig. 29. Oil atomizer

A conventional oil ring, loosely riding on the shaft, cannot be used due to mounting conditions and such a ring would hamper shaft extraction from the casing. Introduction of a simply driven oil pump is connected with the appearance of excessive rubbing parts. Besides, the pump drive would hamper the dismantling procedure.

A reasonable way out is mounting upon the shaft a folding spring-loaded oil atomizer.

The atomizer (Fig. 29) is the lever *3* made of thin-sheet steel and fixed to the front bearing anti-splash disk. Spring *1* holds the lever against the shaft. The action of centrifugal force on the lever

overcomes spring tension, the lever swings out radially and dips into the oil sump. As the pump stops, the spring returns the lever to its initial position back to stop *2*. The shaft may then be easily extracted from the casing.

The slight outbalance arising from the swinging lever is easily eliminated with the aid of a small counterweight *4*, secured on the anti-splash disk.

Since the oil atomizer is a self-adjustable device, the amount of oil is automatically supplied at approximately the same rate regardless of sump oil level. When striking the oil surface, the atomizer is deflected in a direction opposite to shaft rotation, catching each time a small portion of oil, thus precluding excessive bubbling.

The upper oil level in the sump is in line with the lowest points of the ball bearings. With the selected dimensions the total quantity of oil held by the oil sump is approximately 1.3 litre and the working volume, being determined from the atomizer immersion depth (its most advanced position), is about 1 litre. This enables the pump to be run for a long period of time without the addition of fresh oil.

To ventilate the oil cavity a breather is installed, which is also used for oil filling. The most rational place to mount the breather is close to the rear bearing in plane *A-A* (see Fig. 28), far away from the operation zone of the atomizer. For stable readings an oil gauge should also be installed in the same zone.

The breather consists of housing *10* with extension sleeve *3*, which protects the breather from oil splashes. In the housing a long cylindrical gauze filter *4* is placed so that oil can be poured in through a large-size funnel.

The filter is secured to the housing shoulder by washer *8* sliding along rod *7* which, in its turn, is fixed inside breather cap *9* and loaded by spring *6*. On the breather housing the cap is fastened by a bayonet lock and held in position by the same spring *6*.

An assembly of washers *5* mounted on extension of the rod *7*, preclude the throw-out of oil droplets through the breather. When the cap is removed, washer *8* sits on the shoulder of rod *7* and at the same time the washer assembly *5* is taken out, exposing the filter ready for filling.

An oil gauge glass is set on the same side as the breather. This enables the oil level to be controlled during filling. Directly behind the glass with a small space between a white plastic screen is fitted, having holes top and bottom which communicate with the oil sump. The screen has two functions: it shows the oil level and protects the oil gauge glass from splashes, when filling or running.

For the oil cavity to be viewed from the opposite side to the oil gauge a hatch provided with an easily-detachable lid should be made.

To fasten the pump to its mounting frame four screwed holes should be provided: two (*a*) in the breather location plane *A-A*, and the other two—near the discharge volute.

Thus from this stage of design the general arrangement of the pump is drawn (Fig. 30).

The oil drain hole *a* must be positioned in the sloping channel of the oil cavity. To avoid mixing of oil sediments we separate the channel from the atomizing zone by a baffle plate. The drainage

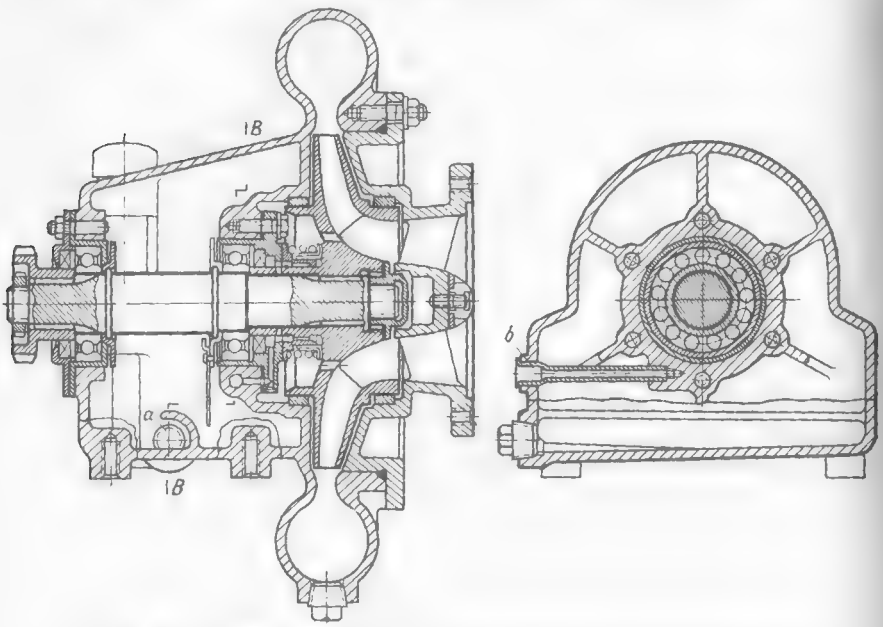


Fig. 30. General view of pump (general arrangement drawing)

tube should be on the same side as the breather and oil gauge glass. The seal drainage tube should also be brought to the same side.

Drainage is effected through tube *b* screwed into the front bearing boss. The opposite end of the tube is flared to fit the casing wall.

(*l*) *Alternative Design. Smaller-Size Volute*

Sketch the pump with the smallest radial sizes, according to Fig. 20*d*. This time the impeller is given a conical form (Fig. 31) and the volute is offset to bring it closer to the pump casing.

The drainage channel of the face seal must be inclined and also shifted to the side to bypass the volutes.

In this version it is better to fasten the pump directly to the housing of the flange-mounted drive electric motor using adapter *1*.

With such a fastening it is no longer necessary to install the pump on a frame. In addition, the drive clutch will be totally enclosed by the adapter casing, thus the installation as a whole will be more compact and lighter.

The design is quite suitable for fitting to a flange-mounted electric motor. The addition of fastening holes in the pump base for

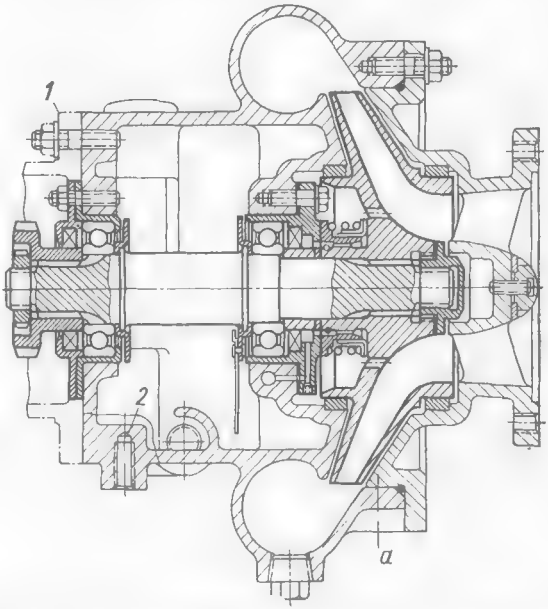


Fig. 31. Pump with smaller volutes (general arrangement drawing)

attachment to a frame, if necessary, will make the fastening still more universal.

If so, the front fastening holes should be made in the pump casing flange (plane *a*) leaving the back holes (*2*) in place.

Considering the smaller-size volutes as a positive advantage of the construction we establish this as the main variant. The initial version (see Fig. 30) is also brought forward for discussion, as well as the individual pump units sketched during the composition (e.g., automatic water discharge unit, Fig. 23, which may not be necessary).

(*m*) *Durability*

In addition to the measures accepted earlier, assuring bearing durability, we introduce induction hardening of the shaft surfaces where the bearings are fitted, making the hardness of these surfaces

not less than 50 Rc. Finally rolling with hardened rolls to still further improve the hardness. The shaft is made of steel grade 45.

To prolong service life of the lubricant and bearings, oil stipulated in specifications and containing stabilizing additives is used (e.g., complex additives ЦИАТИМ-330, АЗНИП-8).

Pump durability depends primarily on the service life of face seals and the corrosion-resistance of the impeller, the pump casing and other parts in contact with water.

Seal durability is determined by the material of the rubbing surfaces. The stationary casing seals are made of stainless steel grade 4X13, improved by nitriding (700-800 VPH). The rotating disk seal is made from the same steel but its working surface is coated with a composite layer of cermet bronze-graphite, impregnated also with silicone plastic.

The following materials are suitable for manufacture of the impeller and casing:

high-strength grey cast iron grade СЧ 28-48. The strength (in modified state) σ_b = 26-30 kgf/mm^2, hardness, 180-250 BH, specific weight, 7.2 kgf/dm^3. The material casts well. Its disadvantages are brittleness (elongation $\delta < 0.3$ per cent) and comparatively low corrosion resistance in water;

corrosion-resistant cast iron grade ЖЧНДХ 15-7-2 (*Ni-resist*). Tensile strength σ_b = 25 kgf/mm^2, hardness, 150-170 BH, specific weight, 7.6 kgf/dm^3. This material advantageously differs from grey iron by a higher plasticity (δ = 3 to 4 per cent) and its corrosion resistance in water is 15-20 times higher;

silumin grade АЛ4 (8-10% Si; 0.4% Mn; 0.25% Mg; balance—Al). Its strength (in modified state) σ_b = 15-25 kgf/ mm^2, hardness, 70-80 BH, elongation δ = 2-3 per cent, specific weight, 2.65 kgf/dm^3. The material has good casting properties. Its corrosion resistance in fresh water lies between those of grey cast iron and Ni-resist.

The main advantage of silumin is its low specific weight which offers (with equal sized sections) sharp decrease in stresses (almost three times) when under the action of centrifugal forces in comparison with the other listed materials. However, one should always remember of its lower resistance to abrasion, which is the result of its low hardness. This disadvantage is critical for an impeller subjected to the action of rapid water flows and moving with even greater speed relative to the water layers in the gaps between the casing walls and impeller disks.

From the comparative analysis of the listed materials it is clear that it is better to choose silumin for the casing and Ni-resist for the impeller. The higher price of the latter is repaid in higher durability and reliability.

When building the pump casing of silumin, the softness and plasticity of this material should be kept in mind. Studs should be

used as fasteners and nuts provided with washers. The holes for a drainage plug and pump fastening bolts must have reinforcing threaded steel bushes. Owing to the low rigidity of silumin, casing walls must be at least 8 mm thick and be properly ribbed.

To further corrosion protection of the water cavity walls a zinc protector *1* (Fig. 32) is introduced at the hub of the stationary vane apparatus.

All other parts coming into contact with water (some sealing details, impeller captive nut, drainage plug, etc.) are made of stainless steels: the sealing spring, fastening elements, nuts and drainage plug — from heat-treated steel grade 4X13 and stoppers — from mild stainless steel grade 1X18H9.

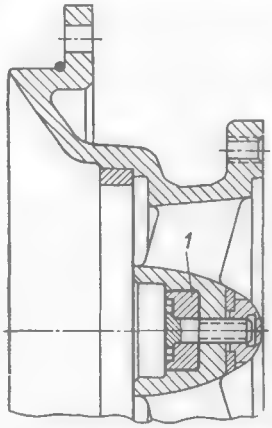


Fig. 32. Installation of zinc protector

Amidst other methods applied to enhance durability and reliability is the heat-treatment of shaft splines, as well as all fastening elements. The splined rim of the drive flange should have a hardness of not below 55 Rc which can be achieved by induction hardening. The surfaces on which the sevanite sealing cups work must have a hardness of not lower than 45 Rc and a surface finish not worse than class 10.

Nuts of internal fastening elements must be locked positively (e.g., by tab washers).

(*n*) *Working Arrangement*

After discussion the final design is chosen and the working arrangement drawn, which is then used for designing the components.

The drawing of the working arrangement of the pump, (see Fig. 33) should contain the main coordination, coupling and overall dimensions, as well as the dimensions of connection and locating joints, classes of fits and grades of accuracy, and code numbers of bearings. The maximum and minimum sump oil levels and sump capacity should also be specified.

The drawing should also indicate the main specifications of the pump (capacity, head, speed, rotation direction, power consumption,

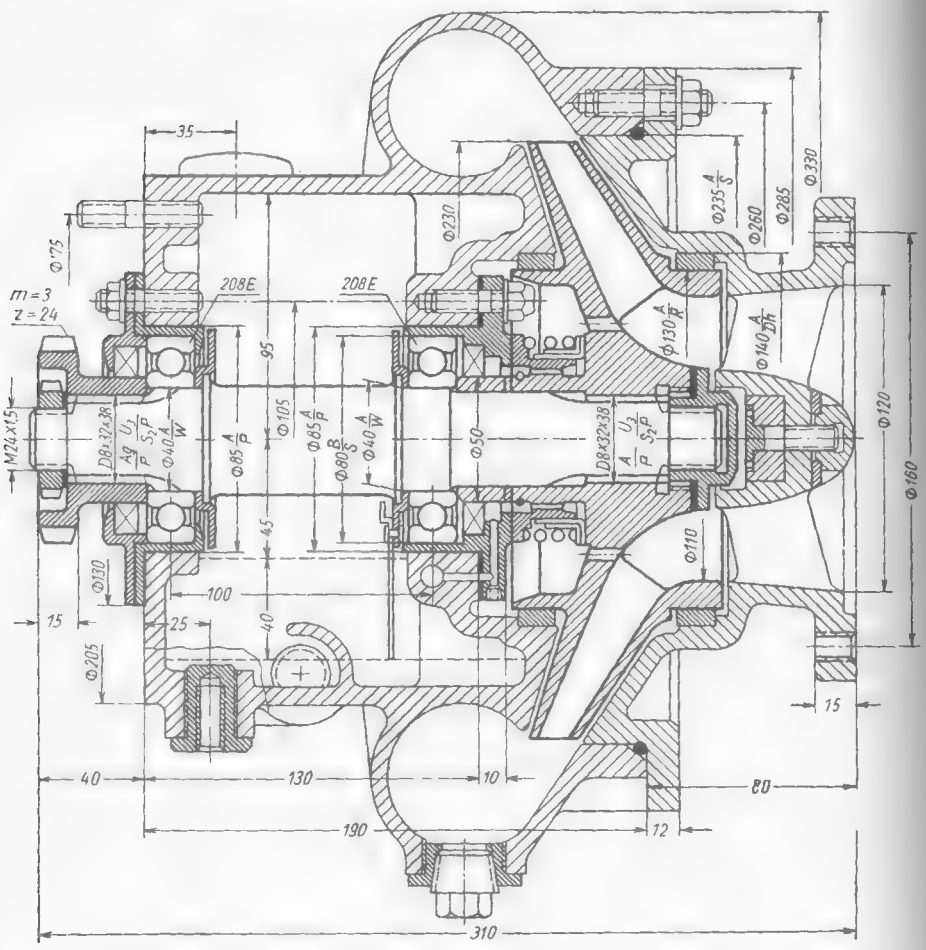


Fig. 33. General view of pump (design drawing)

type of electric motor) and technical requirements (hydraulic pressure testing of water cavities, overspeed test of impeller).

On the basis of the working arrangement check calculations of strength are carried out.

Chapter 3

Weight and metal content

Weight is an important machine parameter, especially in transport and, even the more so, in aviation, where each kilogramme of excess weight lessens the payload, speed and cruising range. In general engineering a reduced weight of machines means metal economy and cheaper production.

The reduction of weight becomes a particularly important factor under mass production conditions, as it enables metal to be saved on a national scale. This in no way relieves us of the necessity for reducing the weight of individually produced machines or those produced in small quantities as their total output makes up in general a considerable fraction of all machine production.

Of course, weight reduction is by no means the end in itself. The cost of material is an inappreciable part of the total sum of expenditures spent during machine exploitation, as expenditure depends primarily on machine reliability and durability. The tendency to reduced weight should be rejected without hesitation if it degrades such machine parameters as strength, rigidity or reliability, especially so in general engineering. It is better to have a somewhat heavier machine than a lighter one possessing worse reliability characteristics and durability.

Comparative weight characteristics of machines of similar purposes are assessed by their weight factor which is the quotient of the machine weight G by the main machine parameter.

For machine power generators such a main parameter is their power N. The weight-to-power ratio of such a machine will be

$$g = \frac{G}{N}$$

This characteristic accounts for the design perfection of a machine, as well as the use of light alloys and non-metallics.

For various types of internal combustion engines the weight-to-power ratios are as follows:

stationary, 8-15; ship, 3-8; automobile, 2-5; aircraft, 0.5-0.8 kgf/e.h.p.

In transport the weight factor is expressed as the ratio between the total machine weight and the payload, and has the following values: for ship transport, 20-30; railway transport, 10-20; automobile transport, 3-5; aircraft, 1.2-2.5.

The quality of metal-cutting machines in these terms is assessed as the ratio between the machine weight and the nominal power of its drive motor (this is not expressive as it neglects the degree to which the nominal power is utilized and, also, the machine productivity).

The design perfection of speed reduction units is evaluated by the weight-to-torque ratio or by the ratio between their weight and the product of the transmitted power and the gear ratio (reduction ratio).

The notion of the metal content must be differentiated from that of the weight. These are not equivalent.

Let us explain this by an example. Assume that two machines have identical dimensions and parameters but one of them is made mostly of heavy metals (steel and cast iron) and the other, of light alloys (e.g., aluminium). It is clear that the weight of the second machine is less than the weight of the first, as the specific weight of the heavier material is approximately twice that of the light alloy, but the metal content, which represents the amount of the metal used, is the same for both machines.

Metal content can best of all be expressed by the volume of metallic parts making up the machine. Then, along with the weight factor, it is possible to introduce an index of the specific metal content (specific volume), expressed as the quotient of the volume of the metal parts by the machine main parameter.

This index will allow the evaluation of first, the economy of metal achieved in a given machine, and second, the quality of the design, i.e., the rationality of the design scheme and the form perfection of the components regardless of the specific weights of the materials used.

As machines are generally manufactured of metals with different specific weights, then, in a general case, the index of the specific metal content will be

$$v = \frac{V}{N} = \frac{\frac{\sum G_1}{\gamma_1} + \frac{\sum G_2}{\gamma_2} + \ldots + \frac{\sum G_m}{\gamma_m}}{N}$$

where $\sum G_1, \ldots, \sum G_m$ = total weight of components made from materials with respective specific weights $\gamma_1, \ldots, \gamma_m$

N = machine main parameter

The reciprocal N/V can be defined as the volume utilization factor.

A reduction in weight along with a decrease in the metal content may be obtained by giving parts rational cross-sections and shapes, by reasonably utilizing the strength characteristics of materials, by making use of stronger materials and rational design schemes, by eliminating excessive safety margins and substituting non-metallics for metals.

3.1. Rational Sections

Maximum reduction in weight can be achieved by giving machine elements complete equal strengths. The ideal case is when the longitudinal stresses are distributed equally in every cross-section and at every point in the cross-section. This is the condition when equal strength is most fully expressed. Here, the material is used to its best possible advantage, and the weight of a part, for the designed stress level, is the minimum obtainable. In principle, such a case is possible only for a few types of loading when the load is taken by all cross-sections of a part, that is in tension and compression and, partially, in shear.

In flexure, torsion and in complex states of stress the stresses along the section are distributed unevenly. They achieve their maximum at the extreme points of the section and at other positions may fall to zero (e.g., on the neutral axis of a section subjected to flexure). In these cases it is only possible to approximate conditions of equal strength by equalizing the stresses over the entire section, removing metal from the less loaded portions and adding it at the most loaded points, i.e., at the periphery.

Let us take, for example, a cylindrical component subjected to flexure or torsion (Fig. 34).

The stresses in a solid component part having a continuous round section (implied here are ordinary bending stresses and torsional shear stresses) are distributed according to the law of a straight line (Fig. 34*a*) passing through the section centre (in the Figure the stress diagram pertaining to the case of flexure is turned through 90° from its true direction).

Removal of metal from the weakly loaded centre zone, i.e., making it tube-shaped, assures a more equal stress distribution in the remaining zones (Fig. 34*b*). The thinner the wall, i.e., the greater the d/D ratio, the more regular is the stress distribution. Keeping the external diameter constant causes the wall stress level to rise. However, increasing the outside diameter easily reduces the stress level to the previous values (Fig. 34*c*) or even significantly decreases it (Fig. 34*d*).

This principle, which can be defined as the principle of equally stressed sections, may be applied to sections of any shape.

The weight savings obtainable under such circumstances are illustrated in Fig. 35. The latter shows a number of profiles with identical cross-sectional areas and, consequently, with equal weight

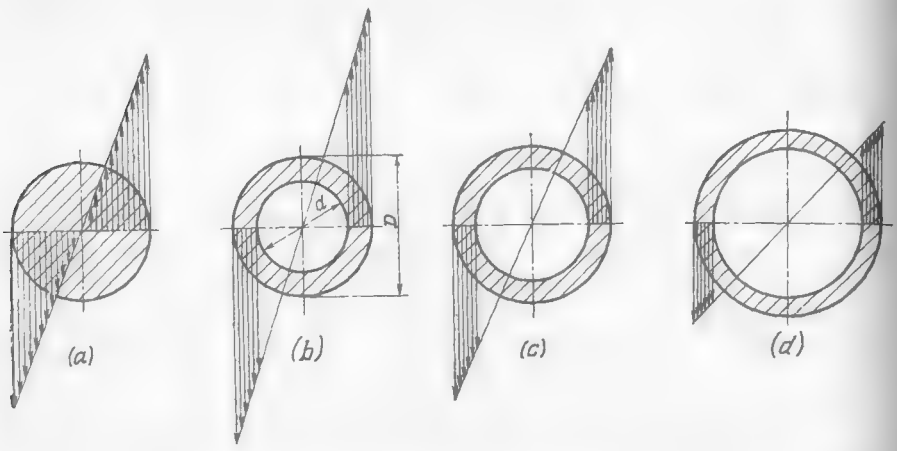


Fig. 34. Distribution of stresses over sections of solid and hollow cylindrical components (in flexure and torsion)

per unit metre, in the order of decreasing metal thickness over the periphery. The most rational forms of sections (hollow and rolled steel sections) are noted for high strength and rigidity. The advant-

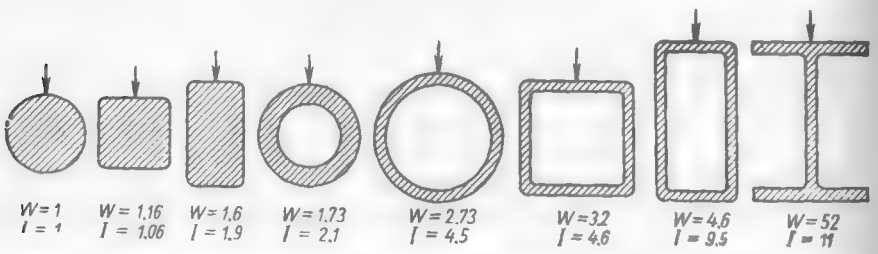


Fig. 35. Resisting moment W and moment of inertia I for solid and hollow stock having identical cross-sectional areas (in flexure)

ages increase with symmetrical shapes proportionally to the increase in radial dimensions entailing under the given conditions ($F =$ $=$ const) wall thinning, but with asymmetric profiles more according to the metal mass concentration in the area of the greatest acting normal stresses (e.g., in rolled section profiles with an increase in the thicknesses and widths of flanges).

(a) *Strength and Rigidity Indices of Profiles*

The relative weight advantage of a profile subjected to bending is characterized by the quantities $\frac{W}{F}$ and $\frac{I}{F}$ (*reduced profile strength and rigidity*). The reciprocals $\frac{F}{W}$ and $\frac{F}{I}$ are defined accordingly as the *reduced profile weight* in terms of strength and rigidity.

These indices have linear dimensions $\left(\frac{W}{F}\ \text{cm},\ \frac{I}{F}\ \text{cm}^2\right)$ and totally define the profile's advantage in shape and linear dimensions.

If the denominator in the expression $\left(\frac{W}{F}\ \text{cm}^3/\text{cm}^2\right)$ is raised to the power $^3/_2\left[\frac{\text{cm}^3}{(\text{cm}^2)^{3/2}}=1\right]$, and in the expression $\left(\frac{I}{F}\ \text{cm}^4/\text{cm}^2\right)$, to the power $2\left[\frac{\text{cm}^4}{(\text{cm}^2)^2}=1\right]$, we shall obtain the following dimensionless expressions

$$w=\frac{W}{F^{3/2}} \tag{3.1}$$

$$i=\frac{I}{F^2} \tag{3.2}$$

which define the degree of rationality of the profile shape regardless of its absolute dimensions. Although rather approximate representations, the expressions (3.1) and (3.2) enable one to have a sufficiently clear impression of the weight advantage.

The dimensionless reciprocals characterize the profile weight

$$g_W=\frac{F^{3/2}}{W} \tag{3.3}$$

$$g_I=\frac{F^2}{I} \tag{3.4}$$

With equal success we may use the following dimensionless indices

$$w'=\frac{W^{2/3}}{F} \tag{3.5}$$

$$i'=\frac{\sqrt{I}}{F} \tag{3.6}$$

or

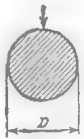
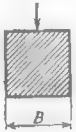
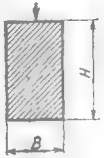
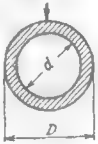
$$g'_W=\frac{F}{w^{2/3}} \tag{3.7}$$

$$g'_I=\frac{F}{\sqrt{I}} \tag{3.8}$$

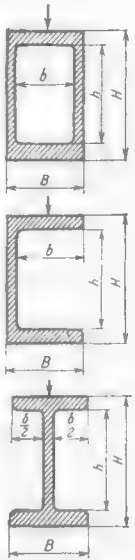
Some actual values for the indices w and i, calculated from formulae (3.1) and (3.2) for the most widely used profiles (flexure in the vertical plane), are given in Table 4.

Table 4

Specific Indices of Strength and Rigidity of Profiles (in Flexure)

Sketches of profiles	F	W	I	$w=\frac{W}{F^{3/2}}$	$i=\frac{I}{F^2}$
	$0.785D^2$	$0.1D^3$	$0.05D^4$	0.14	0.08
	B^2	$B^3/6$	$B^4/12$	0.166	0.083
	B^2c $(c=H/B)$	$B^3c^2/6$	$B^4c^3/12$	$0.166\sqrt{c}$	$0.083c$
	$0.785\,D^2(1-a^2)$ $(a=d/D)$	$0.1D^3(1-a^4)$	$0.05D^4(1-a^4)$	$0.14\,\frac{1-a^4}{(1-a^2)^{3/2}}$	$0.08\,\frac{1-a^4}{(1-a^2)^2}$
	$B^2(1-e)$ $(e=b/B)$	$\frac{B^3}{6}(1-e^4)$	$\frac{B^4}{12}(1-e^4)$	$\frac{1-e^4}{6(1-e^2)^{3/2}}$	$\frac{1-e^4}{12(1-e^2)^2}$

Continued

Sketches of profiles	F	W	I	$w = \frac{W}{F^{3/2}}$	$i = \frac{I}{F^2}$
	$BH(1-e\eta)$ $(e = b/B;\ \eta = h/H)$	$\frac{BH^2}{6}(1-e\eta^3)$	$\frac{BH^3}{12}(1-e\eta^3)$	$0.166\,\frac{1-e\eta^3}{(1-e\eta)^{3/2}}$	$0.083\,\frac{1-e\eta^3}{(1-e\eta)^2}$

(b) Strength and Rigidity of Round Hollow Profiles

Round profiles (shafts, axles, etc.) are of particular importance in mechanical engineering. Let us consider some typical cases which illustrate the advantage of hollow profiles under conditions of flexure and torsion.

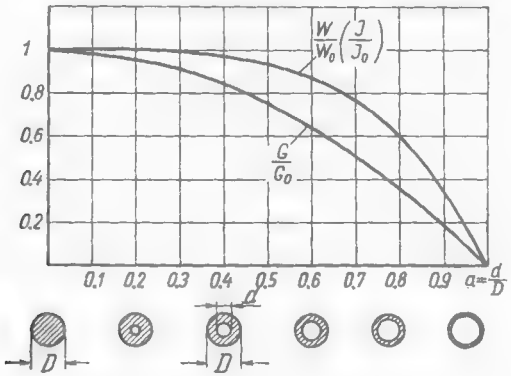


Fig. 36. Change of resisting moment W, inertia moment I and weight G of cylindrical components as a function of $a = \frac{d}{D}$ with D = const (in flexure and torsion)

Case 1. Given: outside diameter of a component part (D = const). For such a case the following relations are valid:

relative strength and rigidity

$$\frac{W}{W_0} = \frac{I}{I_0} = 1 - a^4$$

relative weight

$$\frac{G}{G_0} = \frac{F}{F_0} = 1 - a^2$$

In these formulae the subscript 0 refers to a solid round section, and the value a is the ratio between the hole diameter d and the outer diameter D of the part $\left(a = \frac{d}{D}\right)$.

Assume W_0, I_0 and G_0 of a solid part to be equal to unity. Then in Fig. 36 we can show changes in the resisting moment, moment of inertia and weight of the part with the increase of the ratio a.

From this figure we see that:

small-diameter holes ($d < 0.2D$) hardly affect the strength, rigidity and weight of the part;

when a = 0.3-0.6 a significant reduction in the weight with a simultaneous reduction of the strength and rigidity are observed

(when $a = 0.6$ the weight of the part is almost 40% less, while the strength and rigidity are only worse by approximately 10%).

Thus, in the case being considered it is possible to introduce holes with diameter $d = 0.6D$ and obtain a large weight reduction with an inappreciable reduction in the strength. Increasing the values of d in excess of $0.6D$ considerably lowers the strength.

Case 2. Given: the shaft strength ($W = \text{const}$). The outside diameter of the part changes.

For this case the following relationships are valid

$$\frac{D}{D_0} = \frac{1}{\sqrt[3]{1-a^4}}$$

$$\frac{G}{G_0} = \frac{F}{F_0} = \frac{D^2}{D_0^2} = \frac{1-a^2}{(1-a^4)^{2/3}}$$

$$\frac{I}{I_0} = \frac{1-a^4}{(1-a^4)^{4/3}} = \frac{1}{\sqrt[3]{1-a^4}}$$

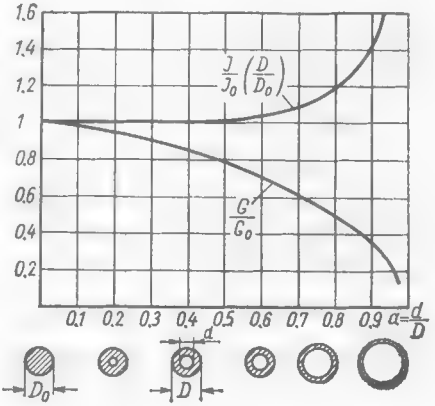


Fig. 37. Change of inertia moment I, outside diameter D and weight G of cylindrical components as a function of $a = \frac{d}{D}$ with $W = \text{const}$ (in flexure and torsion)

Figure 37, plotted on the basis of these relations, shows the rigidity and weight indices of a part as a function of $a = d/D$.

With d/D and D increasing simultaneously, the weight and rigidity characteristics of a part continuously improve. The enlargement of the outside diameter, a condition for equistrength, is hardly noticeable at the beginning. Even when $d/D = 0.7$, the O.D. of a part must only be increased by 10%, then the weight is lowered by 40%. The moment of inertia will increase with the same regularity as the outside diameter.

The regularities shown in Fig. 37 are further expressed in Fig. 38, where cylindrical components having equal strength in flexure and torsion with progressively ascending d/D ratios are shown.

After a certain point the d/D ratio cannot be increased as the wall thinning may result in local deformation, particularly in areas where loads are applied, making difficult also the use of such design elements as keyways, recesses, threads, etc. Actually $d/D > > 0.7$ is very rarely used. Parts in which $d/D = 0.8$-0.95 refer to pipes, tubes and shells.

Thin-walled tubes, having walls 1-2 mm thick, can be used to good advantage for torque transmission, provided significant longitudinal and transverse loads are absent. The design elements which take up torque are welded to gimbal drives of the tubes. Automobile cardan shafts are such examples.

At large d/D ratios the saving in weight may be considerable; e.g., when $d/D = 0.95$ the weight of a tube is as small as 20% of an equally strong solid shaft and its torsional rigidity is almost twice as high as that of the solid shaft.

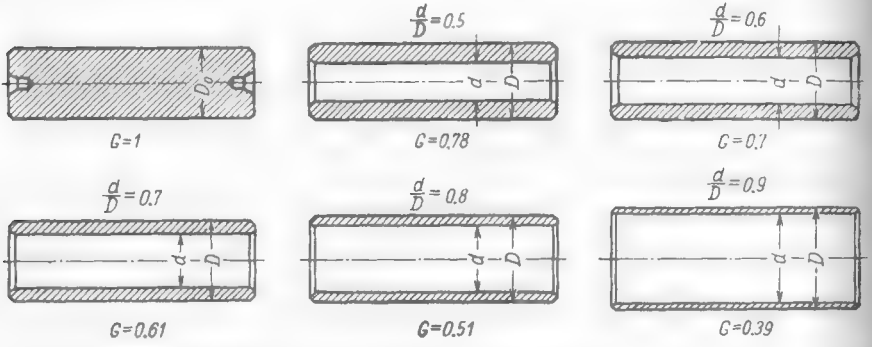


Fig. 38. Cylindrical components of equal strength in flexure and torsion, having different ratios $a = \frac{d}{D}$

Case 3. Given: the shaft weight (G = const).
The calculation formulae for this case are

$$\frac{D}{D_0} = \frac{1}{\sqrt{1-a^2}}$$

$$\frac{W}{W_0} = \frac{1-a^4}{(1-a^2)^{3/2}}$$

$$\frac{I}{I_0} = \frac{1-a^4}{(1-a^2)^2}$$

The values of D/D_0, calculated by these formulae, are illustrated in Fig. 39 as a function of $a = d/D$.

The graph illustrates the advantages of hollow thin-walled sections. When $d/D = 0.9$ the resisting moment and the moment of inertia are increased 4.5 and 10 times respectively, and when $d/D = 0.95$—6 and 20 times respectively when compared with a solid part of the same weight.

Increasing the relative sizes of the outside diameter in parallel with the simultaneous introduction of hollow shell sections and holes leads to a sharp increase in the strength and rigidity indices, which is also accompanied by reduction of weight, giving improved operational characteristics to shafts and parts associated with them as well as providing safe margins which allow machine power and speed to be enhanced.

In modern high class machines solid shafts are almost always replaced by hollow ones.

The regularities considered in this section underlie the modern mechanical engineering trend to applying, whenever possible, thin-walled, tubular and shell constructions for components when extremely high strength and rigidity in conjunction with minimum

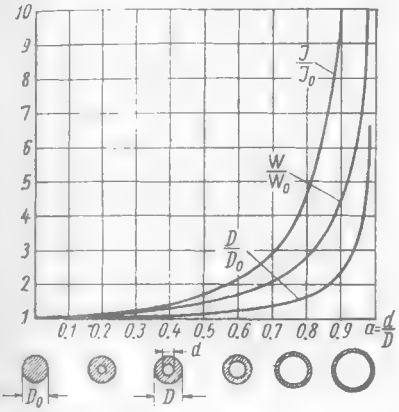


Fig. 39. Change of resisting moment W, inertia moment I and outside diameter D of cylindrical components as a function of $\alpha = \frac{d}{D}$ with $G = \text{const}$ (in flexure)

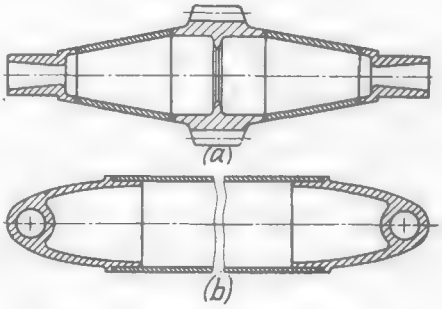


Fig. 40. Shell-type components
(*a*) gear; (*b*) strut

weight are required. Dangerous loss of local stability due to the influence of working loads is prevented by increasing local rigidity at critical points by using tension-compression braces.

Examples of shell-like structures incorporating tubular elements are shown in Fig. 40*a* and *b*. These structures are welded to the solid elements.

(*c*) *Designs of Equal Strength*

In the case of torsion, bending and complex state of stress, when the equality of stresses throughout the cross-section is, in principle, unobtainable, components considered equally strong are those which have identical maximum stresses in each section along the axis.

In bending, a design is considered to be equistrong if the ratios of the bending moment acting at each given section to the resisting moment of the given section are identical. For torsion this condition consists in the equality of the torsional resisting mo-

ments of resistance across each given section, and in cases of complex states of stress, in the equality of the safety factors.

The concept of an equistrong design is applicable both to some parts and to a machine as a whole. Of equal strength are those constructions in which the parts have identical safety factors with respect to the loads acting on them. This rule can be applied to parts made of different materials. Thus, a steel part with stress 20 kgf/mm^2 and yield limit $\sigma_{0.2} = 60$ kgf/mm^2 will be equistrong with an aluminium part having stress level equal to 10 kgf/mm^2 and yield limit $\sigma_{0.2} = 30$ kgf/mm^2. In both these cases the safety factor is 3. This implies that the two parts will succumb to plastic deformation as soon as the loads acting upon them are tripled. Independent of this, each of the parts being compared may still possess equal strength in the above-said meaning, i.e., have identical stress levels along their entire length.

The value of the working loads and stress levels at various sections of a part are found by calculation. A part already calculated as equistrong, will indeed be equistrong if the calculations correctly determine the true values and stress distribution along the part axis, which is not often the case.

Shapes required by equal strength conditions are sometimes technically difficult to obtain and must be simplified. Additional elements on almost all parts are unavoidable (trunnions, shoulders, grooves, recesses, threads, etc.); sometimes they cause local strengthening, but more often, stress concentrations and weakening of the part which also means the introduction of corrections for a true stress distribution along the part.

For all these reasons the concept of equal strength is rather relative. In practice design of an equistrong part amounts to an approximate shape reproduction, dictated by equistrength requirements, with necessary measures taken to reduce sources of local stress concentrations.

Weight savings from the application of the equal strength principle mostly depend on the type of loading and the method of accomplishing equal strength. Some idea of the weight saving can be conceive from the example below where a cylindrical part, to be of equal strength when supported at its ends, is subjected to flexure by a centrally applied transversely acting load (Fig. 41).

Case 1. The part is made equistrong by changing its external shape.

Maximum normal stress in the central section of the original cylindrical part *1* (Fig. 41*a*)

$$\sigma_0 = \frac{M_0}{0.1D^3}$$

where M_0 is the bending moment at the beam centre, equal to the product of the end reaction and the distance $L/2$ from the central section to the reaction plane.

Stress in any arbitrary section

$$\sigma = \frac{M}{0.1D^3}$$

where $M = M_0 \frac{2l}{L}$ is the bending moment in the given section;

l = distance from the given section to the end reaction

Consequently

$$\sigma = \frac{2M_0 l}{L0.1D^3}$$

The maximum stress in any section of an equistrong designed part

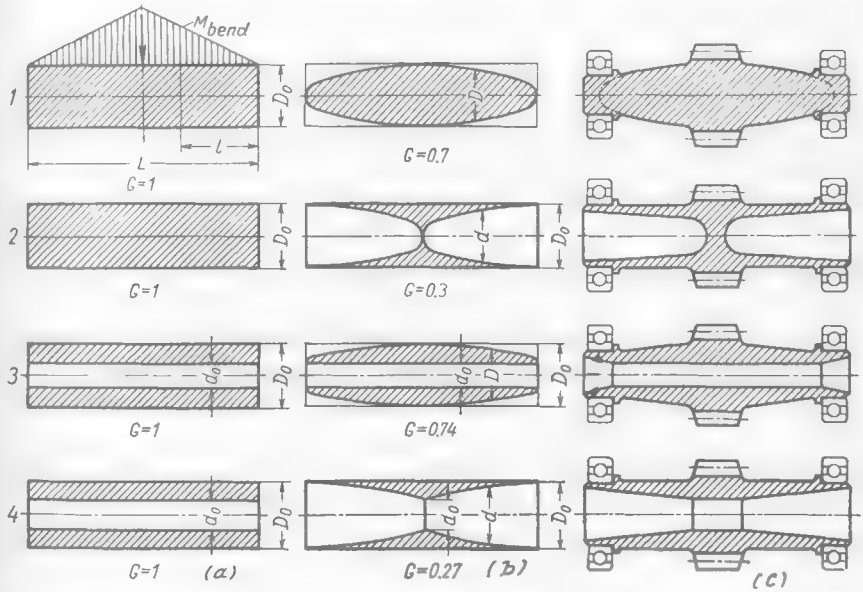


Fig. 41. Methods of imparting equal strength properties to cylindrical parts (in lateral flexure)

(a) original shapes; (b) shapes of equally strong parts; (c) design versions of equally strong parts

must be constant and equal to

$$\sigma = \sigma_0 = \frac{M_0}{0.1D^3} = \frac{2M_0 l}{L0.1D^3} = \text{const}$$

hence, the diameter for an equistrong part should be

$$D = D_0 \sqrt[3]{\frac{2l}{L}}$$

The profile of the equistrong part *1* is shown in Fig. 41*b*. Figure 41*c* illustrates a design of equistrong part *1* for a gear made integral with a shaft mounted in antifriction bearings.

The equistrong shapes are more simple. The simplified shaft has bearing journals at its ends.

Case 2. Part *2* is made equistrong by removing internal metal from the shaft while keeping the outside diameter unchanged.

The equistrength requirement

$$\sigma = \sigma_0 = \frac{M_0}{0.1D_0^3} = \frac{2M_0 l}{L0.1D_0^3(1-a^4)} = \text{const}$$

where a is the ratio between the varying diameter d of the internal lightening recess and the constant outside diameter D_0.

Hence, the current internal diameter will be

$$d = D_0 \sqrt[4]{1 - \frac{2l}{L}}$$

The profile of the equistrong part *2* in this case is as shown in Fig. 41*b*, and its design, in Fig. 41*c*.

The high weight saving (weight of the equistrong part is one third of that of the original) is the result of applying not only the equistrength principle, but also the principle of equal cross-sectional stress.

However, it should be noted that this method of imparting equal strength means enlargement of the bearing support diameters which naturally decreases somewhat the resultant weight saving.

Case 3. Equal strength of a hollow part *3* (Fig. 41) is obtained by changing its outer configuration.

The equistrength requirements for this case are expressed by the following formula for determining the varying outer diameter:

$$D = D_0 \sqrt{\frac{1-a_0^4}{1-a^4} \cdot \frac{2l}{L}}$$

where $a_0 = d_0/D_0$ is the initial ratio between the hole and outside part diameters;

a = current value of d_0/D for the equistrong part

With moderate values of a_0 the weight saved in this case will be close to that in the case of part *1*.

Case 4. Equal strength of hollow part *4* (Fig. 41) is achieved by changing its internal shape.

The value of the varying diameter of the internal cavity satisfying the equal strength requirements is

$$d = D_0 \sqrt[4]{1 - \frac{2l}{L}(1-a_0^4)}$$

where $a_0 = d_0/D_0$ is the ratio between the hole size and the outside diameter of the original part

The weight saved in this case is similar to that in the Case *2*.

It should be emphasized that other conditions being equal the rigidity of a part having equal strength is inferior to the rigidity of parts which have even locally improved safety factors.

In those cases when the rigidity of a part is important lowering of its value can be averted by lowering the stresses (which, naturally, lowers the weight saving) or by applying in each separate case a rational method of achieving equal strength.

Thus, the equistrong component *2* (Fig. 41*b*) achieved by removing internal metal, is much more rigid than part *1* (Fig. 41*b*), al-

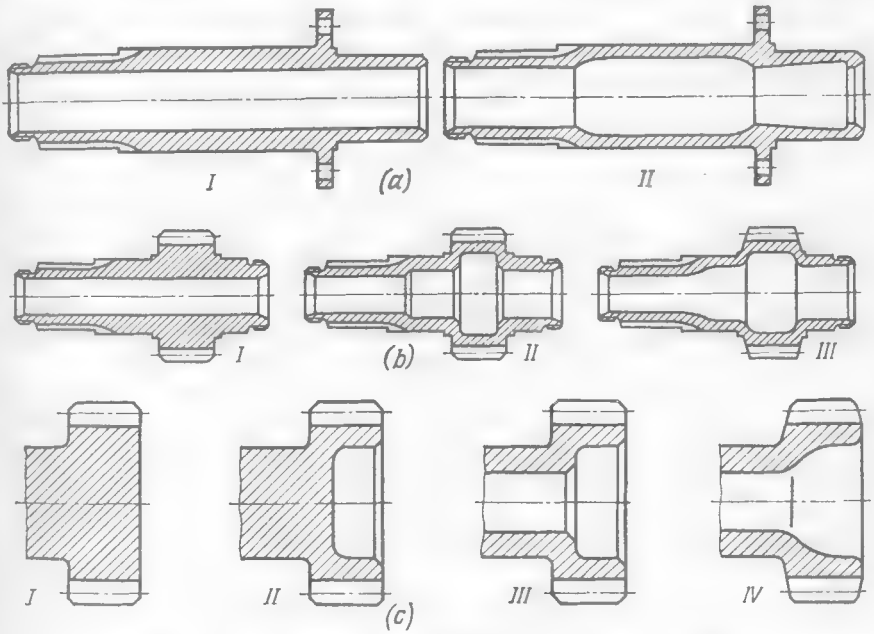


Fig. 42. Imparting equal strength properties to parts

though it is lower in rigidity than the original solid cylindrical part *2* (Fig. 41*a*).

Some examples of imparting equal strength properties to parts are illustrated in Fig. 42.

Flanged shaft *I* (Fig. 42*a*) loaded by a constant torque has unequal strength between the splines and flange. The maximum stresses occur on the splined portion, while between the splines and flange, where the shaft diameter is enlarged, the stresses are considerably less. The equistrong (stress-balanced) design, *II*, given in Fig. 42*a*, has been obtained by calculations, based on the constancy of the torsional resisting moment across any section of the shaft.

The design of pinion-shaft *I* (Fig. 42*b*) with a through hole of constant diameter is of unequal strength in spite of its simplicity and technological effectiveness. Shaft *II* with stepped-turned portions is approximately of equal strength. Design *III* is a carefully considered design aimed at improving fatigue strength and having a smooth shape of the internal recesses.

Shafts *II* and, particularly, *III* are much more expensive to manufacture. Nevertheless, the necessity for lightness and higher fatigue strength often justifies more complicated and expensive production.

A lighter design is exemplified by the terminal gear made integral with the shaft (Fig. 42*c*).

Here, in order to obtain the shape, which approximately meets the equal strength requirements, upsetting is first used followed by machining (variant *IV*).

It is especially important to observe equal strength requirements in discs which rotate at high speeds (turbine rotors, centrifugal and axial compressors). The centrifugal forces developing in such components cause stresses increasing in the inward radial direction (towards the hub) as a result of the summation of centrifugal forces occurring in each radial layer of metal from the periphery to the disc centre. The conditions for equal strength in this case require the disc to be tapered becoming thinner towards the periphery. This measure lightens the disk as a result of removing metal from the periphery and contribute to lesser hub stresses.

Equal strength calculations for rapidly rotating disks are complex since in a number of cases thermal stresses must be considered. These stresses appear as a result of temperature variations over the disk body. In many cases the picture is complicated by the effect of thermal shock produced under certain operating conditions by unstable heat fluxes flowing inwardly or outwardly.

(*d*) *Equal Strength of Units and Connections*

Implementation of the equal strength principle, as applied to units and joints, we consider by examples.

Figure 43*a* shows a turnbuckle tightening two screwed rods. Design *I* cannot be regarded as of equal strength, since elementary calculations prove that the tensile stresses in the tubular section are three times less than those in the rods.

In the equal strength design *II* the turnbuckle cross-section has been reduced.

As a general remark to this example, we should say that circular sections are deceptive when their strength is evaluated visually. The sectional strength of such components is proportional to the square of their diameter the resisting moment and torsion, to the cube, and the moment of inertia, to the fourth power. These conditions are

not always taken into account when designing such components. When evaluating tensile or compressive strengths and also rigidity of round parts the designer generally makes the mistake of exaggerating their sizes.

The design (Fig. 43*b*) of chain conveyer link joints with lugs of the same width is not equistrong for the following three reasons:

the tensile safety margin at the base of the lugs in the upper link is 3/2 times less than that in the lower link (the figure shows the ratio between the numbers of lugs in the upper and lower links);

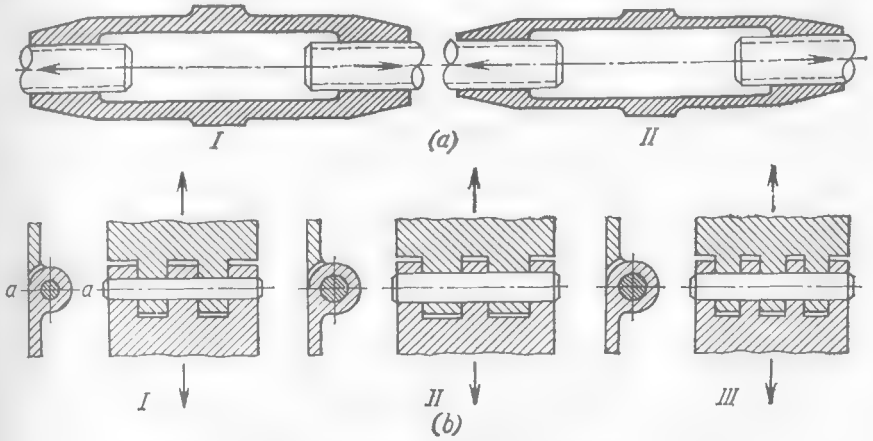


Fig. 43. Imparting equal strength properties to units

the shear safety margin for the pin stud is half the tensile safety margin of the lower link lugs;

the tensile safety margin of the lug heads across section *a-a*, with head wall thickness equal to the lug base thickness, is twice that of the lug bases.

In other words, the lugs of the upper link are weaker than their lower counterparts, and the pin strength is less than that of the whole link. The lug head walls are also too thick.

In the equistrong design *II* the total lug width in the lower and upper links is the same, which assures the given equality of stresses in the lugs. The pin diameter is accordingly enlarged; and the lug wall heads thinned to suit.

In the construction *III* equal strength of the pin has been obtained by increasing the number of lugs (seven in place of five in the previous example). In this way the pin diameter can be decreased by $\sqrt{2/3}$ times as compared to that in version *II*.

3.2. Lightening of Parts

Very often equal strength requirements cannot be met either due to the complex configuration of a part or because of the indeterminancy of the acting stresses which develop in it. Under such circumstances the weight of a part is reduced by removing metal from the less stressed areas.

Examples of part lightening are presented in Figs. 44-47 and in Table 5.

Figure 44*a* shows how the weight of the crankshaft webs can be reduced. The outside corners of webs *I* are not involved in transmit-

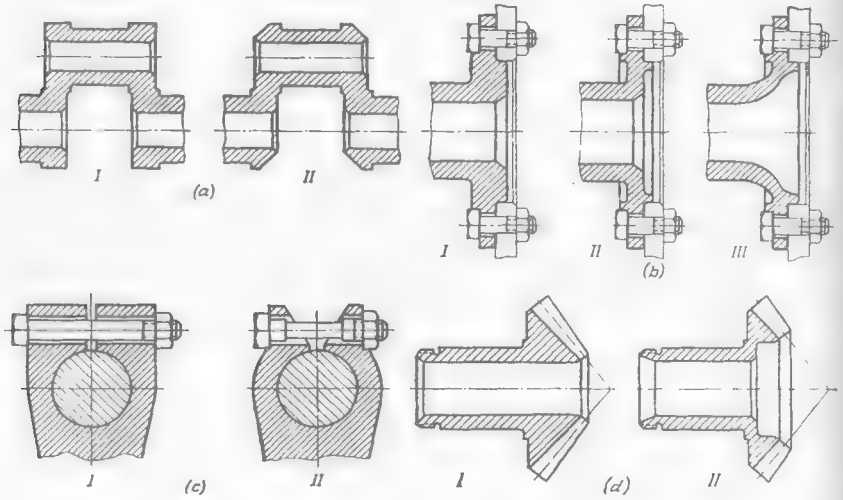


Fig. 44. Examples of lightening parts and joints

(*a*) crankshaft; (*b*) flanged shaft; (*c*) clamp connection; (*d*) bevel gear; *I* — original design; *II-III* — reduced-weight designs

ting forces from crank necks to the crankshaft. Elimination of these web corners (version *II*) substantially reduces the weight without reducing the strength.

The bevel gear (Fig. 44*d*) can be lightened by removing all nonusable teeth portions over the minor diameter. Apart from the weight saved, the shortened teeth assure a more uniform pressure distribution over the tooth length and lessens tooth loads due to the increase in the average pitch diameter.

In flange-type components a substantial reduction of weight can be achieved by changing the flange outside contour (Fig. 45). The radius of the fastening holes centres circle is assumed to be the same in all cases. The weight of the round flange is assumed to be equal to unity.

The possibility of lightening a part by removing surplus metal must never be neglected, no matter how small the amounts may be (Fig. 48). Although weight savings in each separate case may be small the total effect from many similar components may be quite considerable.

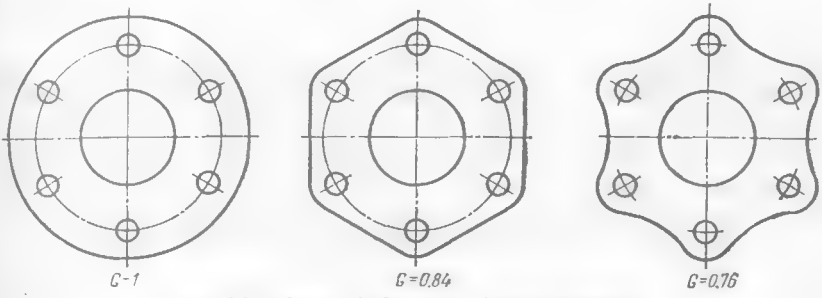


Fig. 45. Effect of shape on flange weight

Particular attention should be paid to decreasing the weight of fastening elements, as this results in an appreciable reduction of the total machine weight. Besides, the use of rationally shaped

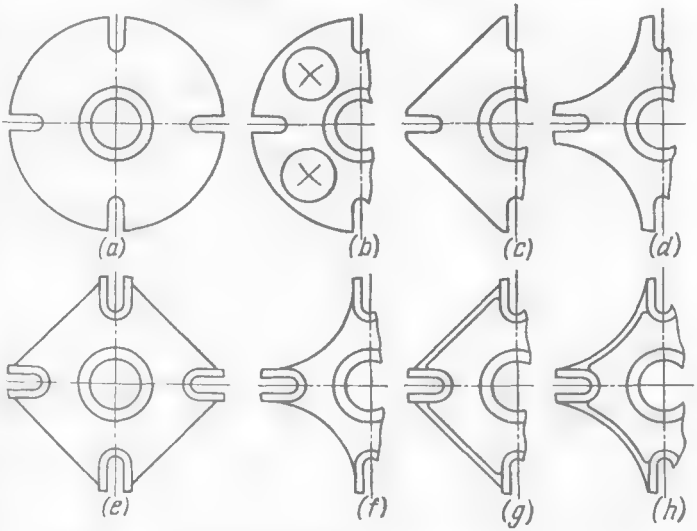
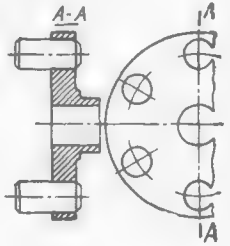
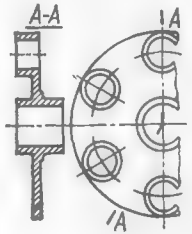
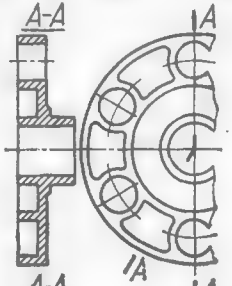
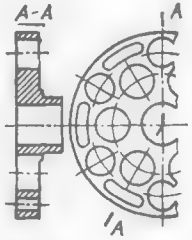
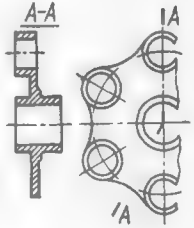
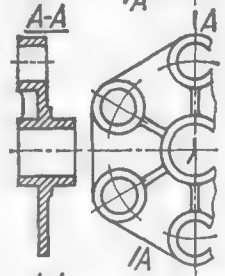
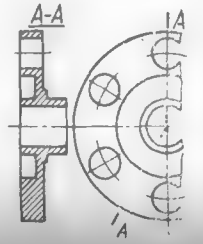
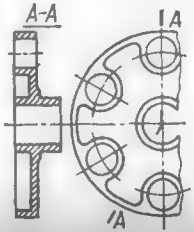
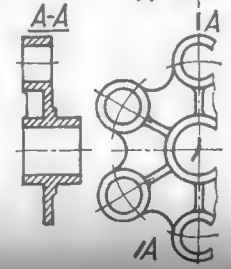
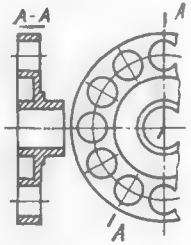


Fig. 46. Methods of lightening a grooved drive plate

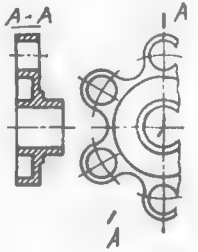
fastening elements is accompanied further by strength and technological advantages. Let us take, for example, the tie bolt shown in Fig. 48*j*. The reduced-weight design, given in sketch *II*, is not only lighter in weight, but also possesses higher resistance to cyclic loads,

Methods of Lightening a Planetary Carrier

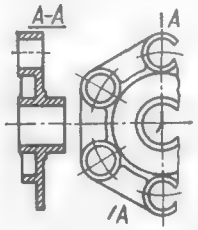
Sketch	Method of lightening	Sketch	Method of lightening	Sketch	Method of lightening
	Original design		Thinning of disk		Fixing flange strengthened by two annular ribs
	Round and elongated through holes		Thinning as above and shaped recesses between fixing holes		Stud hubs and satellite holder connected by radial ribs
	Annular recess between hub and stud fixing holes		To increase rigidity fixing hole hubs have strengthening rib		As above and shaped recess between stud hubs



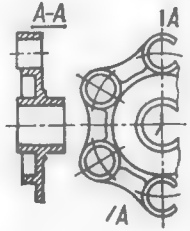
Annular recess as above and extra through holes between fixing holes



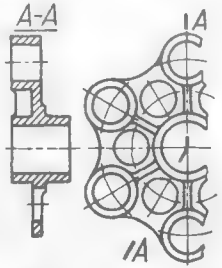
Annular recess as above and shaped recess between fixing holes



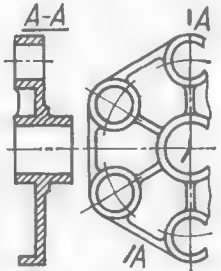
Fixing hole hubs have reinforced with strengthening internal annular rib



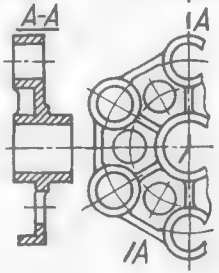
As above and shaped recess between fixing holes



As above and lightening holes in disk



Stud hubs strengthened with peripheral ribs



Peripheral ribs and lightening holes in disk

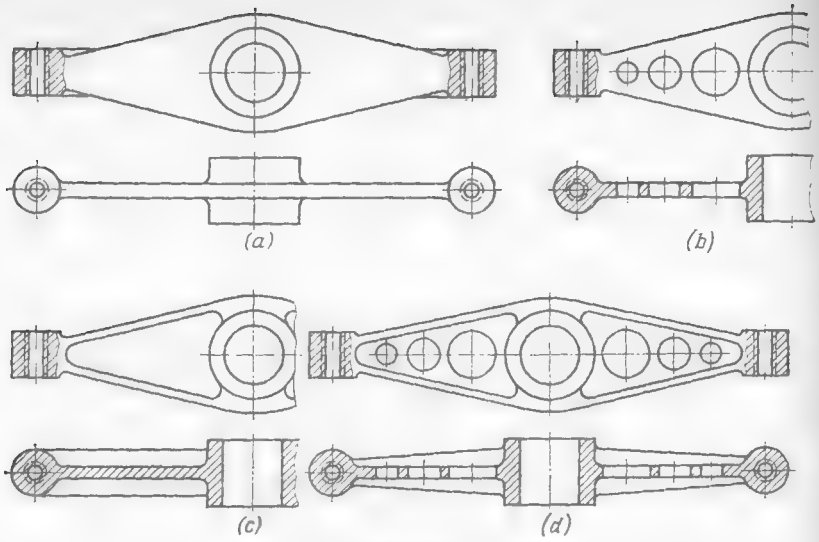


Fig. 47. Lightening of levers

(*a*), (*c*) original designs; (*b*), (*d*) reduced-weight versions

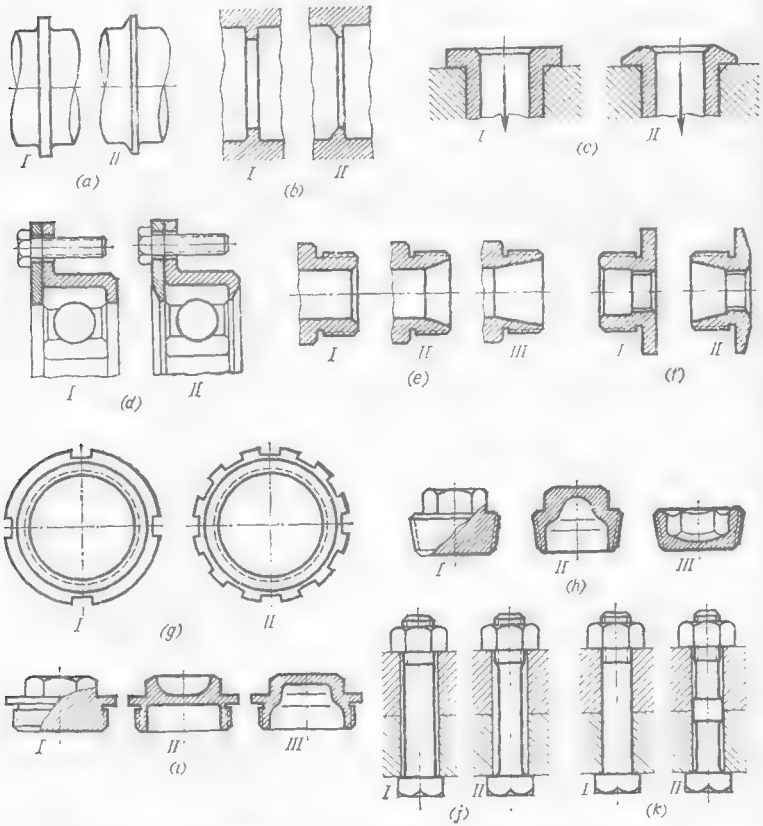


Fig. 48. Methods of lightening component elements

(*a*) shaft collar; (*b*) thrust shoulder in a hole; (*c*) flange; (*d*) disk- and bush-type components; (*e*) threaded shaft end; (*f*) nut; (*g*) ring nut; (*h*), (*i*) screwed plugs and stoppers; (*j*) bolt; (*k*) locating bolt; *I* — original designs; *II* and *III* — reduced-weight versions

particularly if the thread is rolled and the stem coined on a rotary-forging machine.

For the fitted bolt (Fig. 48*k*) the reduced stem diameter will mean less precise machining.

In light-weight machines, where lowered weight plays an important role, the use of reduced-weight nuts and bolts is obligatory.

(a) Effect of Diameter on the Effectiveness of Lightening

When lightening cylindrical components, disks, wheels, rings, as well as parts with figured outlines, e.g., polygons, it should be borne in mind that the greatest effect is achieved when removing material from the periphery and comparatively lesser effect, from parts close to the centre. In the general case the weight saving is proportional to the square of the diameter.

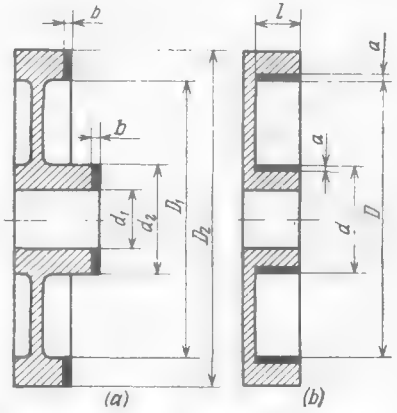


Fig. 49. The weight of part as influenced by:

(a) reduction of axial dimensions in centre and on periphery; (b) removal of annular volumes in centre and on periphery

Let us compare the effectiveness of the weight reduction when decreasing axial dimensions on different diameters.

Fig. 49*a* depicts a disk with a rim and a hub. Let us determine the saving in weight, obtained by removing metal of indentical thickness b from the rim and hub (black portions).

The volume of metal removed from the rim

$$V_1 = \frac{\pi}{4}(D_2^2 - D_1^2)\, b = \frac{\pi}{4} bD_2^2 \left[1 - \left(\frac{D_1}{D_2}\right)^2\right]$$

The volume of metal removed from the hub

$$V_2 = \frac{\pi}{4}(d_2^2 - d_1^2)\, b = \frac{\pi}{4} bd_2^2 \left[1 - \left(\frac{d_1}{d_2}\right)^2\right]$$

The ratio

$$\frac{V_1}{V_2} = \frac{D_2^2}{d_2^2} \cdot \frac{1 - \left(\frac{D_1}{D_2}\right)^2}{1 - \left(\frac{d_1}{d_2}\right)^2} \qquad (3.9)$$

obeys a square relation, relaxed by the effect of the ratio between the outside and inside diameters of the rim and hub.

Assume that the wall thicknesses of the rim and hub are the same and $D_1/D_2 = 0.8$. With the relations given in Fig. 49*a* the value of

$d_1/d_2 = 0.5$ and Eq. (3.9) becomes

$$\frac{V_1}{V_2} = \frac{D_2^2}{d_2^2} \cdot \frac{1-0.64}{1-0.25} = 0.5 \left(\frac{D_2}{d_2}\right)^2 \tag{3.10}$$

With $D_2/d_2 = 3$.

$$\frac{V_1}{V_2} = 4.5$$

In a particular case, when $\frac{D_1}{D_2} = \frac{d_1}{d_2}$ a perfect square relation is obtained

$$\frac{V_1}{V_2} = \frac{D_2^2}{d_2^2} = 9$$

Figure 49*b* illustrates weight reduction by removing annular volumes of the same thickness *a* from different diameters.

In this case the volume of metal removed from the rim will be

$$V_1 \approx \pi Dal$$

The volume removed from the hub

$$V_2 \approx \pi dal$$

The relation

$$\frac{V_1}{V_2} = \frac{D}{d}, \tag{3.11}$$

i.e., it is directly proportional to the ratio of the diameters.

Thus, the comparative saving in weight, obtained by removing metal from different diameters, depends on the mode of relieving and the part shape. Its dependence on the diameter fluctuates within the limits from D/d to $(D/d)^2$.

Radial thinning towards the periphery of disks, flanges, covers, etc., is widely applied, the more so as this shape often conforms to the law of radial stress variation (covers loaded centrally by a transverse force; flanges loaded with a torsional or tilting moment; disks subjected to centrifugal forces, etc.).

The weight saving obtained by thinning disks in the peripheral direction can be evaluated for a most simple case of changing a disk of a rectangular profile (in the meridional section) to most a trapezoidal profile (Fig. 50).

The volume of the square-section disk

$$V_0 = \frac{\pi D^2}{4} B \tag{3.12}$$

The volume of the trapezoidal-section disk

$$V = FS = \frac{\pi D^2}{4} B \frac{1+2\frac{b}{B}}{3} \tag{3.13}$$

The relation between weight G of the trapezoidal-section disk and weight G_0 of the rectangular-section disk will be

$$\frac{G}{G_0}=\frac{V}{V_0}=\frac{1+2\frac{b}{B}}{3}=\frac{2}{3}\frac{b}{B}+\frac{1}{3}, \qquad (3.14)$$

i.e., it is totally dependent on the ratio b/B.

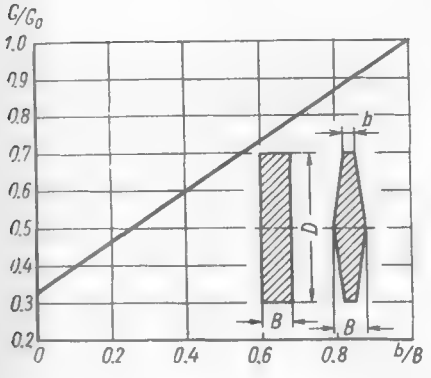


Fig. 50. Relative weight of tapering disks as a function of b/B

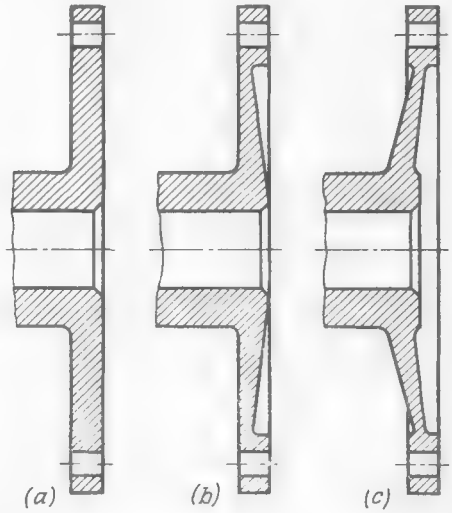


Fig. 51. Tapering of flange towards periphery

When $b/B = 0$ (the case of a triangular profile) the disk will be three times lighter than the rectangular-section disk. For the most widely used ratio $b/B = 0.3$-0.5, trapezoidal-section disk weights will be 0.5-0.65 times less than those of rectangular disks.

A flange thinned towards the periphery is shown in Fig. 51*a*, *b*. To impart better rigidity and stability in cross-sectional planes, the lightened flanges are often made conical (Fig. 51*c*).

Removal of metal from the larger diameters when lightening even small components should be kept in mind.

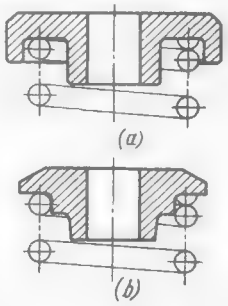


Fig. 52. Centring of spring
(*a*) from outside diameter; (*b*) from inside diameter

For instance, springs should be centred from their inner diameter (Fig. 52*b*) and not from their outside diameter (Fig. 52*a*): this automatically saves weight.

Spacer bushes are better lightened by an external recess (Fig. 53*b*) and not by an internal one (Fig. 53*a*). The weight ratio of the former to the latter (with identical wall thicknesses) is:

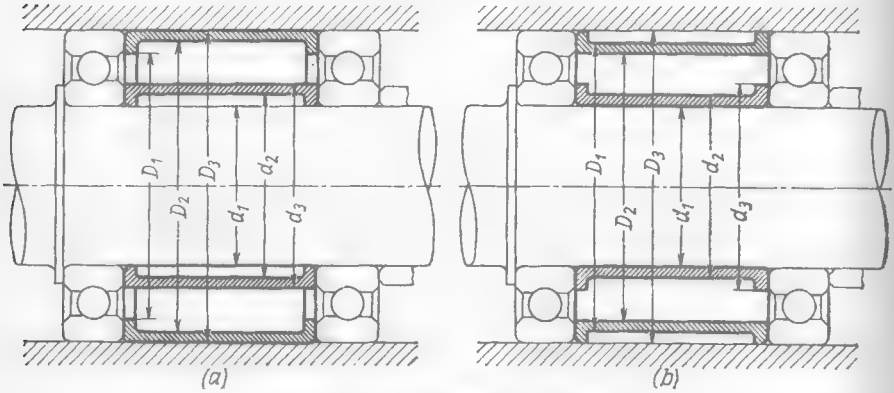


Fig. 53. Lightening of bushes
(*a*) on outside diameter; (*b*) on inside diameter

for the external spacer bush

$$\frac{G_1}{G_2} = \frac{D_2 + D_1}{D_2 + D_3}$$

for the internal spacer bush

$$\frac{g_1}{g_2} = \frac{d_2 + d_1}{d_2 + d_3}$$

and for the relationships given in the Figure, this ratio for the external spacer bush is $\frac{G_1}{G_2} = 0.92$ and for the internal spacer bush: $\frac{g_1}{g_2} = 0.88$. However, although this is a comparatively small weight saving (8-12%), it should not be neglected, if one considers the large numbers of such parts used in mechanical engineering.

(*b*) *Effect of Fillets, Chamfers and Tapers*

The weight of parts can significantly be lowered by increasing wall conjugation radii, i.e., by giving the walls more smooth outlines. The weight savings obtained when changing a right-angle conjugation of walls to a smooth conjugation of radius R and when increasing this radius can be evaluated by the following figures.

Case 1. Conjugation of two flat walls at an angle (Fig. 54*a*). The weight saved by enlarging the fillet radius is easily deduced from Fig. 54*a* by the relation

$$\frac{G}{G_0} = \frac{\pi}{4} \cdot \frac{1}{1 - \frac{r}{R}\left(1 - \frac{\pi}{4}\right)}$$

where r and R = are respectively the original and increased fillet radii;

G and G_0 = respective weights of conjugated portions

From this expression it is clear that the weight saving increases with the increase of R. For a right-angle conjugation ($r = 0$).

$$\frac{G}{G_0} = \frac{\pi}{4} = 0.785 ,$$

i.e., the weight saved with respect to any right-angle conjugation of the same length approximately equals 20%.

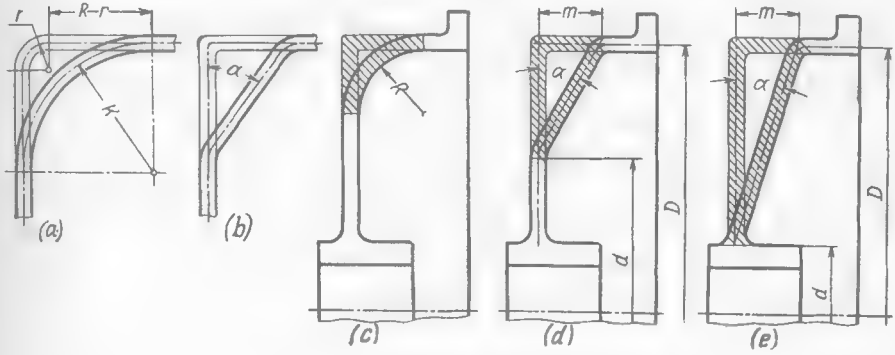


Fig. 54. Lightening of conjugations

Economy of the same order can be obtained by making the wall conjugation skew (Fig. 54*b*). The ratio between weight G of the skew conjugation and right-angle joint weight G_0 is

$$\frac{G}{G_0} = \frac{1}{\sin\alpha + \cos\alpha}$$

This expression has its minimum when $\alpha = 45°$

$$\frac{G}{G_0} = \frac{1}{\sin 45° + \cos 45°} = 0.71$$

Thus, the weight saved, relative to the right-angle joint, is approximately 30%.

Case 2. Conjugation of three mutually perpendicular flat walls. Increasing the spherical fillet radius from r to R gives

$$\frac{G}{G_0} = \frac{1}{\left(\frac{r}{R}\right)^2 + \frac{6}{\pi}\left[1 - \left(\frac{r}{R}\right)^2\right]}$$

For a conjugation at a right solid angle ($r = 0$)

$$\frac{G}{G_0} = \frac{\pi}{6} = 0.52,$$

i.e., the weight reduction is 48%.

For the spatial conjugation of three mutually perpendicular walls, skewed at 45°

$$\frac{G}{G_0} = \frac{\sin 60^\circ}{3 \sin^2 45^\circ} = 0.57,$$

i.e., the weight saving equals 43%.

It should be emphasized, however, that considered here is only the weight reduction of the conjugated section. The weight saved in terms of the part as a whole will naturally depend on the correlation between the weight of the relevant conjugation and that of the complete part.

The examples illustrated in Fig. 54*c-e* show methods of lightening cylindrical bodies consisting of a flat wall and a rim by introducing fillets and tapers in the conjugation section and by replacing the flat wall with a tapered one.

The weight saving achieved with the fillet (Fig. 54c)

$$\frac{G}{G_0} = \frac{\pi (1 - R/D)}{2 (2 - R/D)}$$

where D is the diameter of the rim.

For a right-angle conjugation (R = 0)

$$\frac{G}{G_0} = \frac{\pi}{4} = 0.785$$

Consequently the weight saving obtained with the filletted conjugation is approximately 20%.

The weight reduction due to the tapers (Fig. 54*d* and *e*) is given by

$$\frac{G}{G_0} = \frac{1}{\cos \alpha + \frac{2}{1 + d/D} \sin \alpha}$$

The G/G_0 ratio is graphically shown in Fig. 55 as a function of angle α for different d/D values.

It is evident from the graph that an angle of 60° gives the most substantial weight saving.

The lowering of the weight in the considered case is in the main due to the shortening of the rim length by m (see Fig. 54*e*). In the event of a preset rim length, the conical form will increase the wall weight in the ratio

$$\frac{G}{G_0} = \frac{1}{\cos \alpha}$$

The increase in the weight is insignificant, approaching 4% when $\alpha = 15°$ and 6% when $\alpha = 20°$. This is why weight is often reconciled to the fact that conical walls strongly improve rigidity of a part.

Conical shapes are not recommended for high-speed rotating parts since for the given examples the centrifugal forces will cause a complex three dimensional flexure, tending to reduce the cone to a flat disk.

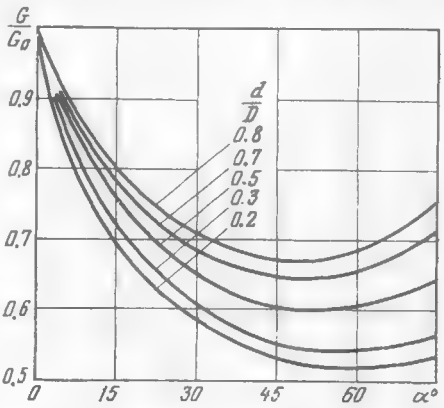


Fig. 55. Relation between the weight of conically conjugated cylindrical parts and angle α

(*c*) *Constructions from Pressed Sheet Metal*

One effective way to save weight is to use pressed metal components (Fig. 56).

The parts, made in the form of solids of rotation and shown in Fig. 56, are manufactured by spinning on lathes (individual or

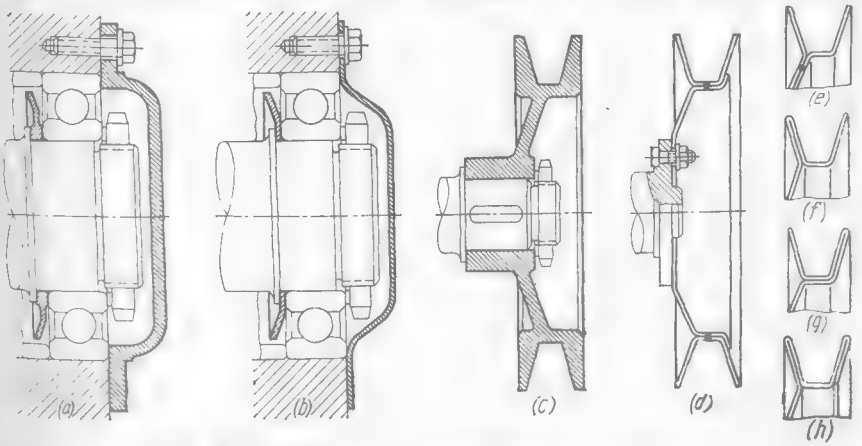


Fig. 56. Substitution of stampings for cast parts

(*a-b*) bearing unit cover; (*e-h*) V-belt pulley

small-lot production) or by pressing. In mass production, when the output scale repays the cost of press tool manufacture, it is certainly economical to produce large-size sheet metal components by pressing (dashes, panels, cases, enclosures, diaphragms, fairings, linings, etc.).

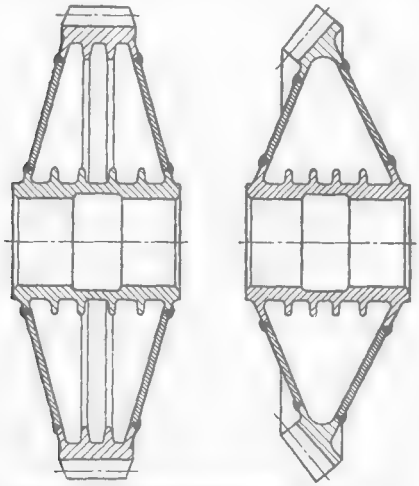


Fig. 57. Gears (shell-type design)

The lower strength and rigidity of thin-sheet parts are compensated for by making them of a shell or arched form, by corrugation and flanging, by introducing ties, braces, and welded stiffeners, etc.

Parts made from plastic metals (low-carbon steels, duraluminum in its annealed or freshly hardened state) with sheet thickness no greater than 3-4 mm are produced by cold stamping, and in the case of deep drawing, in several operations, with intermediate annealing stages. Hot stamping is employed when dealing with sheets thicker than 4 mm.

Some examples of welded thin-walled components are illustrated in Fig. 57.

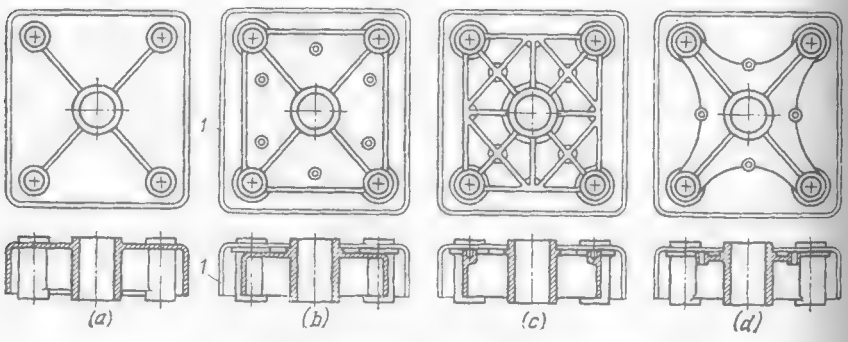


Fig. 58. Table of a housing component

(*a*) one-piece cast structure; (*b-d*) skeleton structures with a thin-sheet skin

In a series of cases substantial reductions in the weight of casings is achieved by using skeleton-constructions (Fig. 58). Elements of parts which must have accurate mutual positions are cast and later connected by lightweight ties. The formed skeleton is then faced with sheet materials.

The original fully-cast structure is shown in Fig. 58*a*. The alternative depicted in Fig. 58*b* shows the first lightening step: the casting is made smaller and the outer, functionally necessary surface, is formed by the thin-sheet facing *1*.

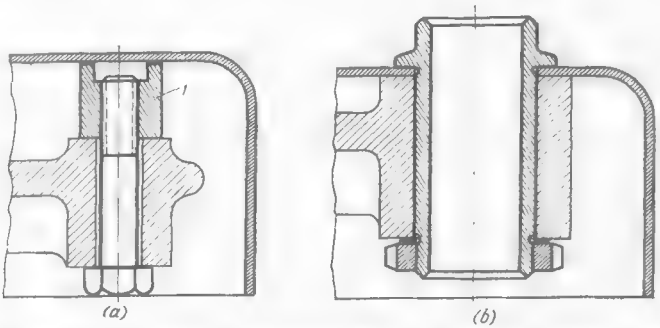


Fig. 59. Methods of securing sheet-metal skin to cast housing components

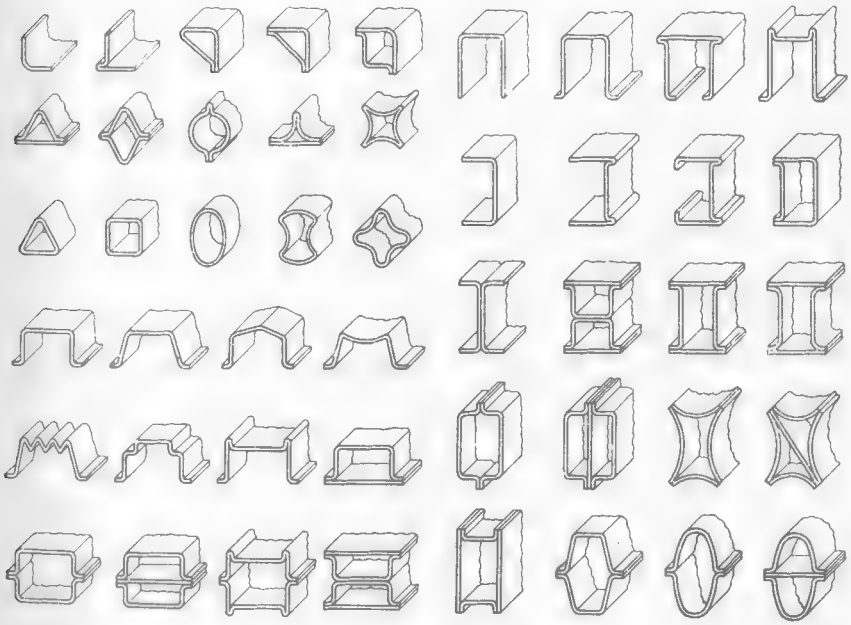


Fig. 60. Reduced-weight shapes

In Fig. 58*c*, the skeleton casting has an open lattice form, which interconnects the central and peripheral bosses; the lattice is covered with a thin-sheet facing.

In another form of skeleton construction, Fig. 58*d*, the central and peripheral bosses are linked together by T-section ribs.

A method of securing steel facings without external fastening elements is shown in Fig. 59*a*. Internally threaded bosses *1* are spot-welded to the facing and afterwards tightly bolted to the casting skeleton.

If the design uses removable bushes, the facing can be secured by tightening the bush flange (Fig. 59*b*).

In particular instances casing components can advantageously be manufactured by welding steel sheets. Such fabrication technique can be applied to simple box-section parts, e.g., gearbox casings. Sheet welded components are much stronger than similar parts made of cast iron. It is not profitable to make intricate casings this way because very many blanks are necessary and a large volume of welding operations.

To save weight frame and girder structures can be replaced to a large extent by sheet steel cold-formed profiles (Fig. 60). Such profiles are generally produced on roll-forming and bending machines.

(*d*) *Extrusion*

A very good way of reducing weight is given by the extrusion process (forcing of a metal, heated to a plastic state, through a die) used these days not only for light alloys but also for steels.

Extruded products can be made of virtually any complex shape to suit a component's function by introducing form mandrels into the die holes. In particular it is possible to obtain profiles rationalized in strength and rigidity with, for instance, internal ribs (Fig. 61*a*, *f*), partitions (Fig. 61*b*, *e*), diagonal ties (Fig. 61*c*), cellular and honeycomb sections (Fig. 61*d*, *g*, *h*).

Of interest is the possibility of obtaining components which have changing profiles along their length. Such shapes are formed by a programmed displacement of the form mandrels relative to the dies due to which the size and contour of the profiting section, through which metal flows out, changes.

When pressing tubes it is possible, by using a stepped mandrel reciprocating to a preset programme, to obtain extended tubes with changing wall thicknesses (Fig. 61*i*), with thicker ends (Fig. 61*j*, *k*), with internal circular ribs (Fig. 61*l*), with wafer ribs (Fig. 61*m*) and even partitions (Fig. 61*n*). Internal helical ribs can be obtained by making the mandrel rotate during extrusion process.

The method is suitable for producing plates with varying thicknesses and rib depths (Fig. 61*o*).

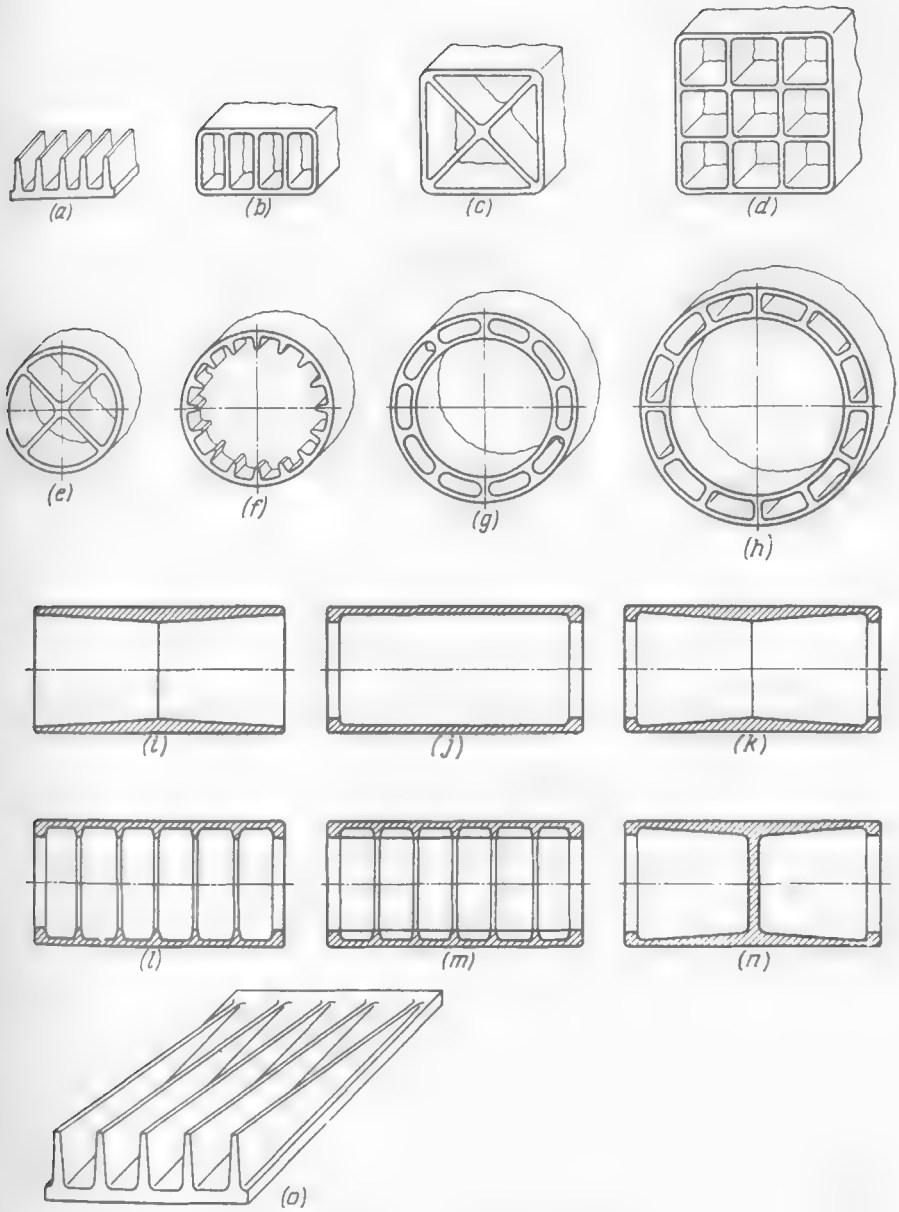


Fig. 61. Extruded shapes

(e) *Effect of Type of Loading*

Rational loading of parts, with the greatest use of the material, is one of the ways of reducing the weight of construction.

Fig. 62 illustrates various ways of loading a round bar. Stress values are shown conventionally by the thicknesses of the full black lines.

In bending (Fig. 62*a*) the most affected portions of the section are found at the section extreme points in the acting force plane. As the stresses approach the neutral axis they become proportionally less

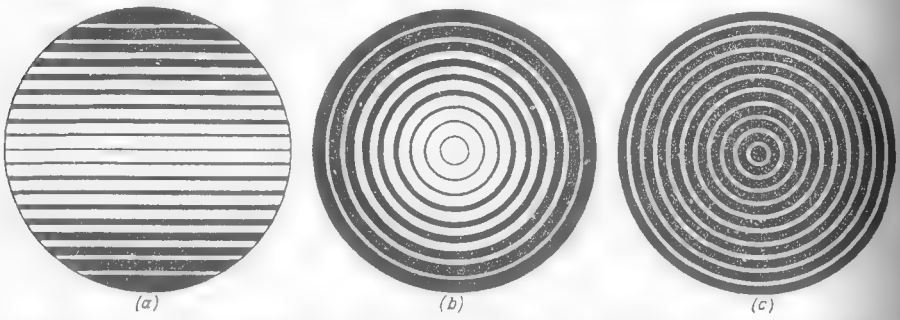


Fig. 62. Stress distribution over the cross-section of a cylindrical part under different loading conditions

(a) flexure; (b) torsion; (c) tension-compression

and are zero at the centre. In torsion (Fig. 62*b*) all peripheral points have identical loads. But the annular stress values decrease and at the centre they are zero.

The most advantageous case is when the material is in tension-compression (Fig. 62*c*) when all points in the cross-section have identical stresses and thus the material is used to its utmost.

Where possible, avoid bending and change to tension-compression stress. The most advantageous constructions as to weight and rigidity are those whose elements work mainly in tension-compression (e.g., girder and rod systems).

If flexure is still unavoidable, for instance, on functional grounds, its negative effects should be suppressed by the following measures:

use rational sections in which the material flow lines are distributed along the lines of maximum stresses (i.e., sections with uniformly distributed stresses);

reduce the bending moment by shortening its bending force arm, i.e., decrease spans, position supports rationally and eliminate cantilever loads which give higher stresses and strains.

Often bending stresses, occurring in a tensile-compression system, are due to asymmetry of sections, off-centre loads or curvilinearity in the shape of the part.

Let us consider an example showing how an eccentrically applied load affects the stresses in a part.

Figure 63*a* depicts a rectangular beam (width a and thickness b) subjected to tensile force P. On one side of the beam there is a recess of width an ($n = 0\text{-}1$). The influence of the recess causes a bending moment equal to the product of force P and the arm $0.5an$.

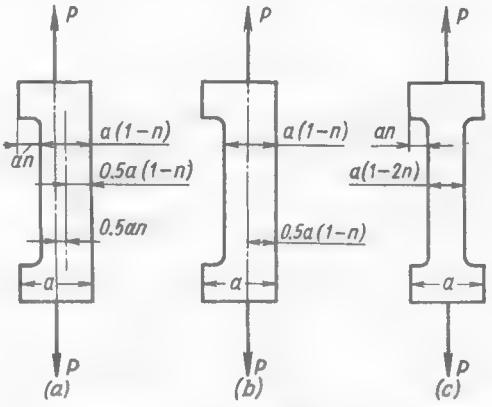


Fig. 63. Determination of stresses for off-centre tension

The maximum tensile stress σ in the beam centre section will equal the sum of the tensile stresses caused by the action of force P and moment $0.5Pan$

$$\sigma = \frac{P}{ab(1-n)} + \frac{0.5 \cdot 6Pan}{ba^2(1-n)^2} \tag{3.15}$$

Assume that the force P is applied at the middle section centre (Fig. 63*b*). In this case the tensile stress in the mid-section will be

$$\sigma_1 = \frac{P}{ab(1-n)} \tag{3.16}$$

Dividing Eq. (3.15) by (3.16), we get

$$\frac{\sigma}{\sigma_1} = 1 + \frac{3n}{1-n}$$

In Fig. 64 the σ/σ_1 ratio is shown as a function of n. From the curve it is clear that the eccentric application of force P increases tensile stresses, their value being directly proportional to the eccentricity. Thus, when $n = 0.25$ the stress is twice as high as in the case when the load is applied at the centre. Hence, such a simple measure as shifting the application point of force P to the centre (in this case by $0.125a$) will reduce the beam stress by half.

Another method of strengthening is the addition of a symmetrically made recess on the opposite side of the beam (Fig. 63*c*). Despite the reduced cross section, the stresses are less due to the elimination of the bending moment. The stress in this case is

$$\sigma_2 = \frac{P}{ab\,(1-2n)} \tag{3.17}$$

Dividing Eq. (3.15) by (3.17) we obtain

$$\frac{\sigma}{\sigma_2} = \frac{1-2n}{1-n} + \frac{3n\,(1-2n)}{(1-n)^2} \tag{3.18}$$

The curve in Fig. 64 plotted on the basis of Eq. (3.18) shows the σ/σ_2 values as a function of n. The introduction of a symmetric bilateral recess with an n value of 0-0.4 ensures a certain gain in

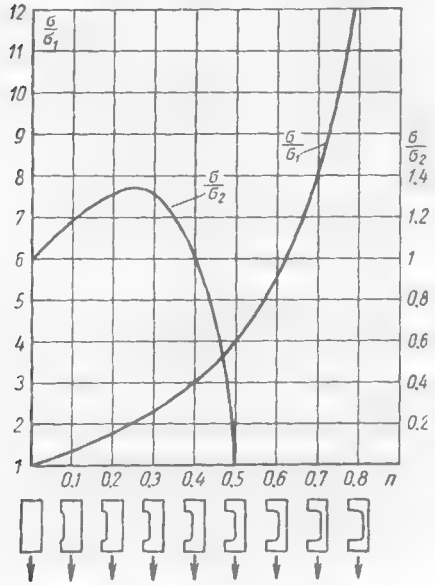


Fig. 64. Stress ratios as a function of recess width n

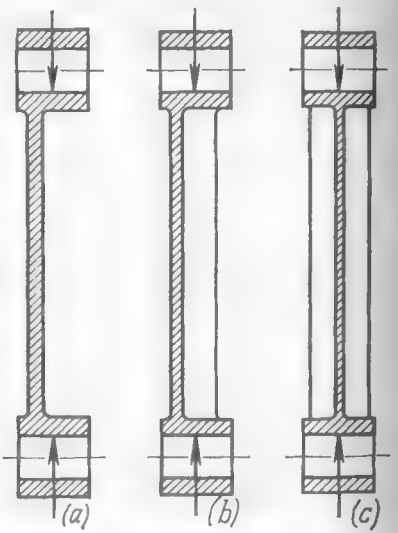


Fig. 65. Relieving connecting rod from bending stresses

strength. When $n=0.25$ and $\sigma/\sigma_2 = \max$, the gain will equal 25%. When $n = 0.4$ the strengths of unilaterally and bilaterally recessed beams are equal.

A connecting rod subjected to a compressive load is shown in Fig. 65. An eccentrically applied load (Fig. 65*a*) produces additional flexural stresses in the rod stem. To sustain these additional stresses the designer has to enlarge the stem section, which, consequently, increases the weight of the unit.

The same drawback, though somewhat less, is encountered in the alternative presented in Fig. 65*b*, where an out-of-centre flexure occurs as a result of the connecting rod stem sectional asymmetry with reference to the direction of the loading force.

A more rational design is given in Fig. 65*c*, in which sections are symmetric in respect to the applied loads. In this case the load is purely compressive. Other conditions being equal, the latter design ensures minimum weight.

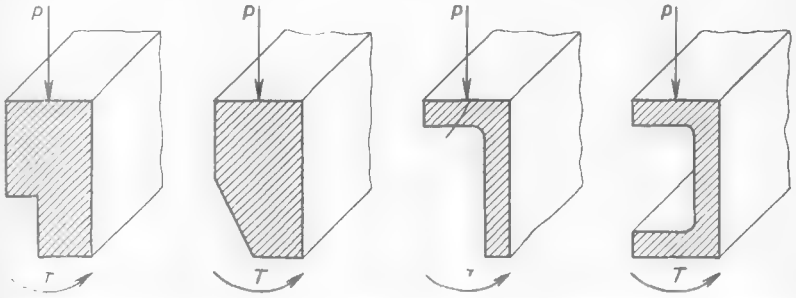


Fig. 66. Shapes of cross-sections causing additional torsion when bending the part by force *P*

In a part subjected to flexure asymmetrical sections cause torsion (Fig. 66) and excessive shear stresses which are summed up with those of flexure.

Another example shows a lever (Fig. 67) with forces applied to its ends in the plane *A-A*. Owing to the displacement of the acting force plane relative to the stem of the lever, the latter is subjected to torsion (Fig. 67*a*, *b*). The correctly designed version, in which the sections are arranged symmetrically relative to the acting forces, is shown in Fig. 67*c*.

When parts are subjected to pure bending, it is advisable to make their sections slightly asymmetric to reduce the tensile stresses on the account of increased compressive stresses.

Most structural materials have better resistance to compression than to tension. Failures almost always begin in those parts of a component which are in tension and not in compression, since the tensile stresses are the first to reveal the internal material defects (microcracks, microflaws, microvoids, etc.). The compressive stresses, on the other hand, make for the closure of microdefects.

This property is very sharply expressed in plastic metals. Figure 68 illustrates tension and compression curves of low carbon steels. Steels under tension pass through the well-known stages: after elastic deformation the metal begins to yield (portion *m*) and, as a result of cold working, hardens (portion *n*). Upon reaching the ultimate

stress limit, a neck begins to form, which terminates in brittle failure of the specimen.

The picture is quite different when the material is subjected to compression. After the elastic deformation period the material continually becomes stronger as a result of both the cold working

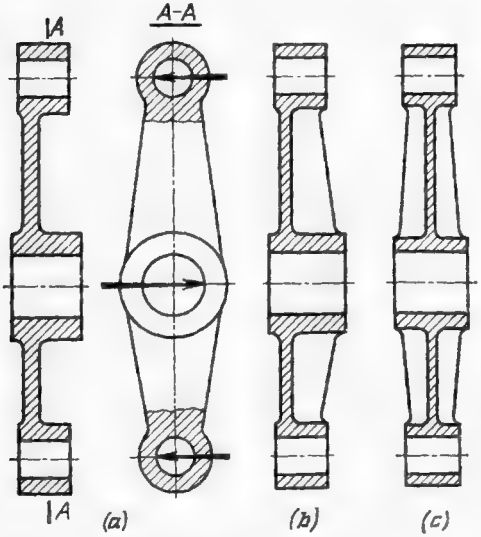


Fig. 67. Elimination of torsion in a lever

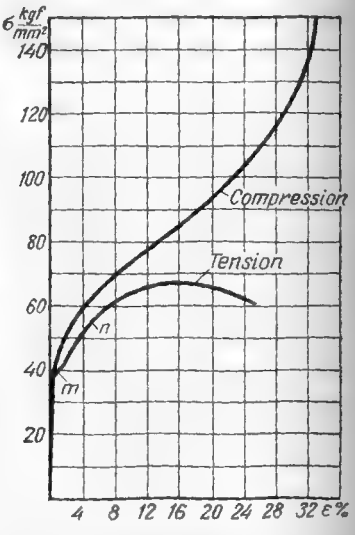


Fig. 68. Loading curves for a constant-diameter specimen made of plastic steel (grade 20) in tension and compression

and transverse expansion of the specimen (cambering). No plastic material can be brought to failure.

In brittle materials (e.g., cast iron) compression results in brittle failure, starting with formation of cracks and terminating in fission. However, such materials display sharp anisotropy of mechanical properties under the effect of tensile and compressive stresses. For instance, the ultimate compressive strength of cast iron is 2.5-4 times greater than tensile strength.

Metals having plasticity somewhere midway between the above described extremes, as a rule offer, better resistance to compressive stresses than to tensile ones. Thus, the ultimate compressive strength of steel grade 45 (hardened and tempered at 100°C), duralumin grade Д16 (after hardening and ageing) and hard brass grade Л-О70-1 exceeds their tensile strength by 1.3-1.5; 1.6-1.8 and 2-2.2 times, respectively.

Exceptions from this general rule are magnesium alloys which, contrary to the above, show better resistance to tensile stresses.

Examples of good and bad loading methods for materials in bending are given in Fig. 69. The lowered level of tensile stresses (Fig. 69*b*, *d*) improves part strength (despite the simultaneous increase of compressive stresses).

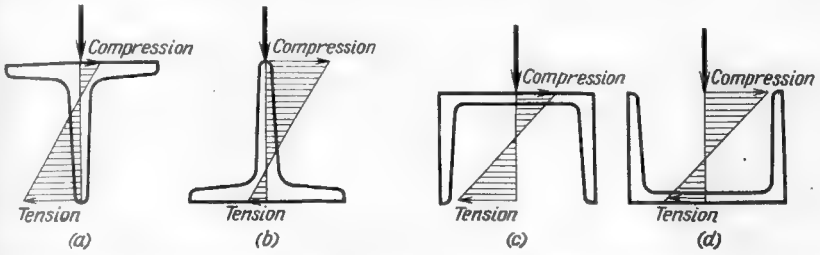


Fig. 69. Loading schemes for asymmetric shapes (stress diagrams are given in the plane of the Figure)
(*a*), (*c*) inadvisable; (*b*), (*d*) advisable

In the constructions presented in Fig. 69*a*, *b* the correlation between maximum compressive and tensile stresses is predetermined by the profile shape and is not always the optimal one.

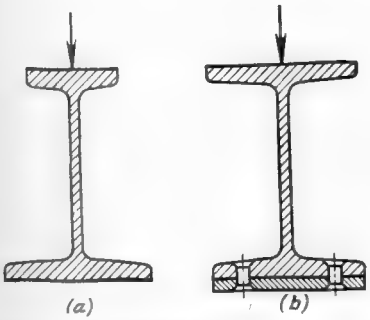


Fig. 70. Reinforcement of sections subjected to tensile stresses

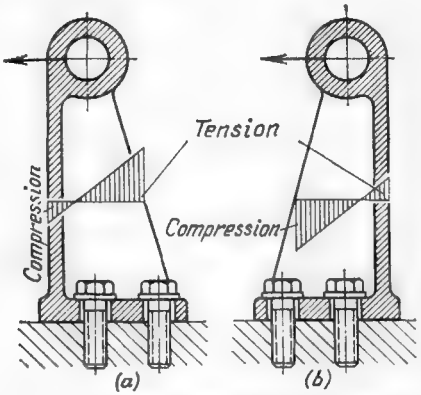


Fig. 71. Design versions of a cast iron bracket
a) poor; (*b*) satisfactory

On the average, the ratio between the admissible compressive and tensile stresses lies in the range of (1.2-1.5) : 1. To utilize this relationship it is better to use slightly asymmetrical profiles, similar to that shown in Fig. 70*a*. The segments subjected to tensile stresses are preferably reinforced with straps made from a material that is stronger than that of the basic part (Fig. 70*b*).

For materials having highly asymmetric strength properties (e.g., grey iron and plastics) with a better resistance to compression, the

relationship between the maximum compressive and tensile stresses is advisably increased in the ratio of their ultimate strengths.

Figure 71 shows irrational and rational designs of a grey iron casting subjected to bending.

3.3. Rational Design Schemes

The greatest possibilities for reducing weight lie in the application of rational design schemes having the least number of parts and the smoothest power arrangement assuring compactness and small constructional sizes.

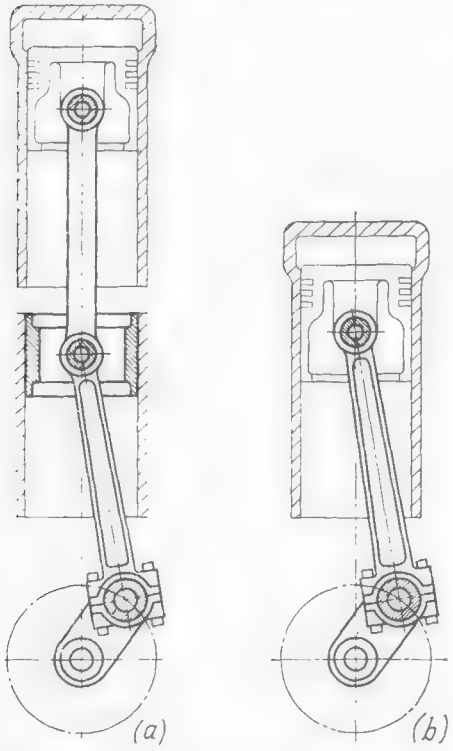


Fig. 72. Elimination of superfluous links in a piston engine

(*a*) *Reduction of the Number of Links*

Elimination of unnecessary mechanism links significantly helps to lower a device's weight. An instance of this is exemplified by the elimination of a crosshead (Fig. 72*a*) in piston engines formerly installed to relieve cylinder walls from lateral forces caused by the connecting rod inclination during the crank rotation. It was proved that the crosshead function could be fulfilled by the piston, if its height were increased and lubrication were better. The height of the trunk piston engines (Fig. 72*b*) has been reduced nearly by half, thanks to the elimination of the crosshead.

A cam drive mechanism is another example (Fig. 73*a*, *b*). In the design shown in Fig. 73*a* the cam actuates the rocker arm through a tappet. In a number of cases it is more rational to let the cam act directly on the rocker arm (Fig. 73*b*). Apart from the lesser number of parts and smaller overall sizes, this scheme brings the acting forces closer together. In the first version the forces are balanced over the casing section h, which means that the casing must be strong enough to withstand the operating forces. In the second version the length of the loaded section h_1 is much less, this enabling a further reduction of the weight.

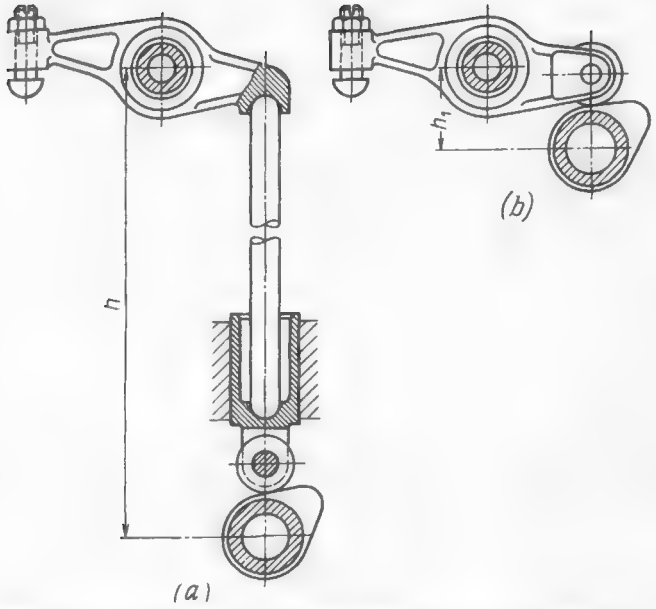


Fig. 73. Elimination of superfluous links in a cam drive

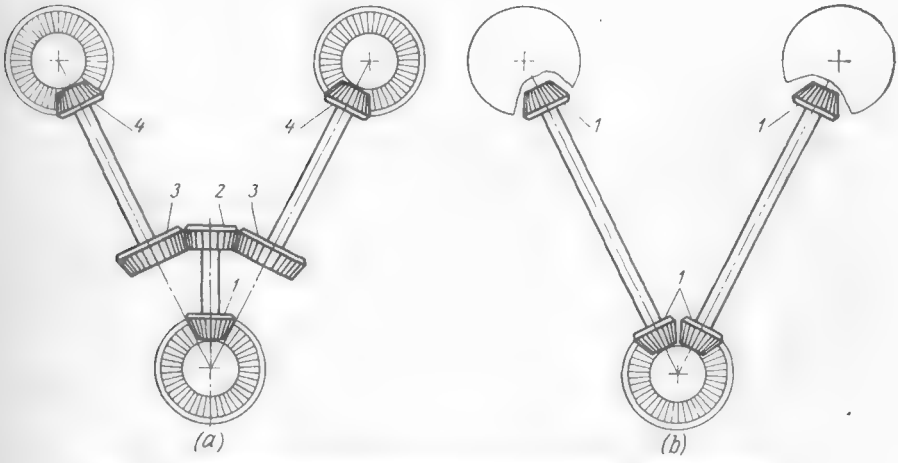


Fig. 74. Elimination of superfluous links in an angle drive

In the unit driving two shafts through a system of bevel gears (Fig. 74) the elimination of superfluous members will lessen the weight of the construction and reduce the number of the gear types from 4 (Fig. 74*a*) to 1 (Fig. 74*b*).

(*b*) *Compactness of Constructions*

Substantial reduction of weight can be reached through rational arrangement of parts and mechanisms which decreases overall size. This is demonstrated by the double-reduction unit in Fig. 75.

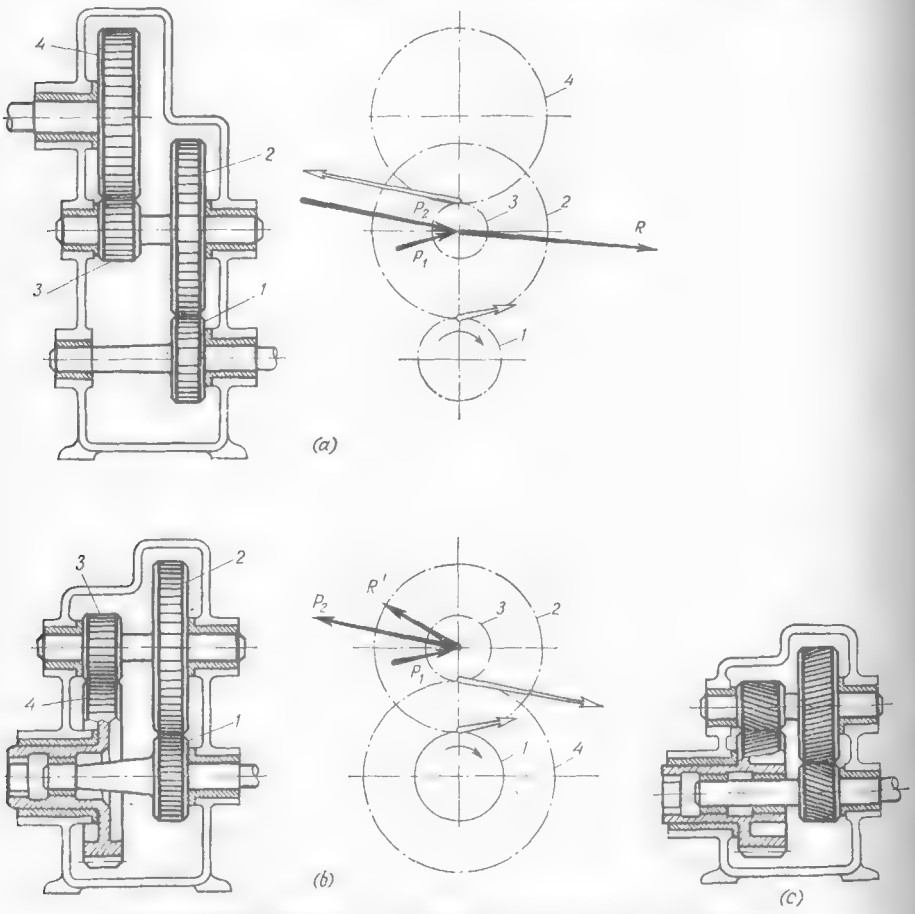


Fig. 75. Reduction of the weight of a double reduction gear

The original design containing a conventional compounded train of gears (Fig. 75*a*) may be made more compact and lighter by mounting the final gear *4* of the train coaxially with gear *1* (Fig. 75*b*).

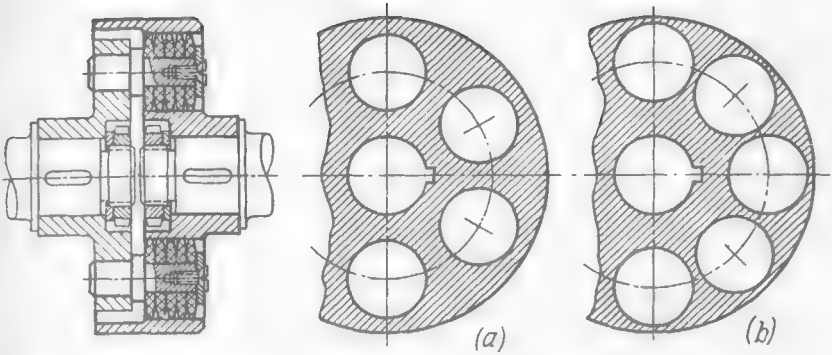


Fig. 76. Rational utilization of the overall dimensions of a flexible coupling

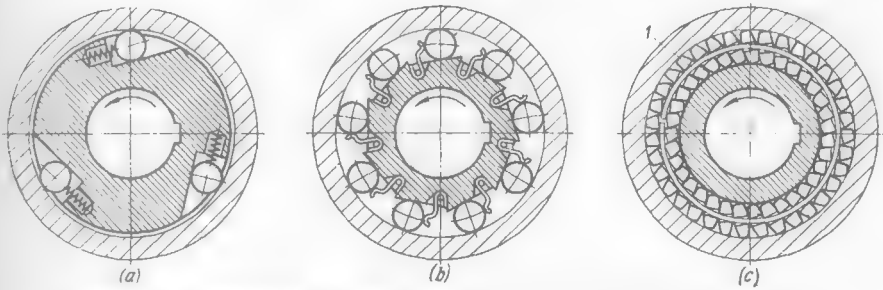


Fig. 77. Increasing the load capacity of an overrunning clutch

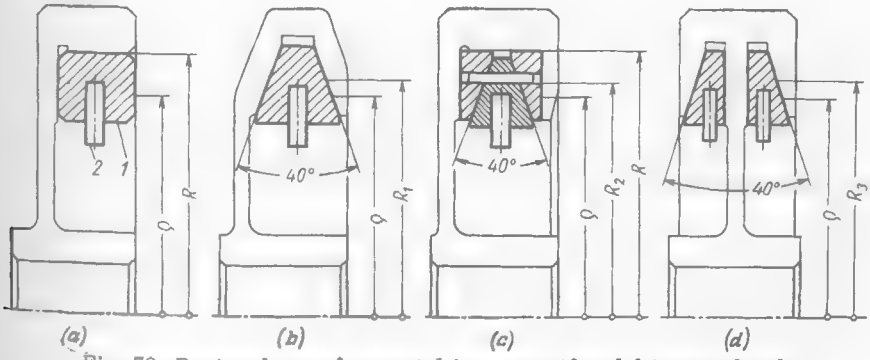


Fig. 78. Rational use of material in a centrifugal friction clutch

Apart from the reduction of the weight and size, such an arrangement of gears *1* and *4* will considerably lower the forces acting upon intermediate gears *2* and *3* and determining the bearing loads, and the loads on the casing walls. In Fig. 75*a* the forces P_1 and P_2 of the drive from the first and the last gears are directed to one and the same side so that the resultant R has rather a large magnitude. In the design shown in Fig. 75*b* the forces act in opposite directions, as a result of which the resultant R' has a much lesser value.

A further reduction of weight and size are obtained by reducing gear diameters (Fig. 75*c*). Increased peripheral forces can be counteracted by increasing the tooth face width, substitution of helical or herring-bone gears for spur gears, use of harder and stronger materials and application of a rationalized lubrication system.

The overall casing size should be utilized to its maximum so that it will accommodate as many operative elements as possible. This principle, which can be described as the principle of close packing, allows significant weight savings to be obtained.

The flexible coupling illustrated in Fig. 76*a* transmits torque through six stacks of washers made from an elastic material. Within the same overall sizes eight stacks of washers can be accommodated (Fig. 76*b*) thus increasing the transmitted torque 1.33 times. Consequently, for a given torque the coupling can also be reduced in weight and size.

A free-wheeling (overrunning) clutch of simple design (Fig. 77*a*) consists of three balls placed into angular cutouts in the driving plate and forced by springs into a jammed position.

The carrying capacity of the clutch can be substantially raised by replacing the balls by rollers (Fig. 77*b*) and increasing their number.

In a still more closely packed construction (Fig. 77*c*) the elements transmitting the torque are in the form of prisms inclined relative to the radial direction at an angle smaller that the friction angle. The tightening split spring ring *1* continuously turns the prisms to a jammed position. In this case the torque is being transmitted by practically the entire periphery of the driving plate. The carrying capacity of such a design is tens times that of the original version.

An example illustrating a rational material utilization is given in Fig. 78 which depicts a centrifugal friction clutch. According to the design alternative shown in Fig. 78*a*, the driving element is a set of annular bronze segments *1* connected by pins *2* with a driver (not shown in the Figure).

The driving force is equal to the product of the total segment centrifugal force $P_{c.f}$ and the coefficient f of friction between the segments and friction surfaces of the driven plate, and is also proportional to the square of the driver's rotational speed (in rpm).

The torque transmitted by the clutch

$$M = P_{c.f} f R = \frac{G}{g} \omega^2 \rho f R$$

where G = total weight of segments, kgf;
g = gravitational constant ($g = 981$ cm/s^2);
ω = angular velocity, rad/s;
ρ = distance from the axis of rotation to the centre of gravity of segments, cm;
R = radius of friction surface (friction radius), cm

In the design presented in Fig. 78 *b* the bronze pieces have a trapezoidal section and are placed in a biconical recess of the driven component. In contrast to the previous version, which has only one friction surface (cylindrical), the second design has two conical friction surfaces.

Owing to the wedge shape of the bronze pieces, the torque transmitted by the clutch, other conditions being equal (i.e., the mass of the pieces and radius ρ remaining the same), will be increased $\frac{1}{\sin \alpha/2} \cdot \frac{R}{R_1}$ times, where R_1 is the new value of the friction radius.

Let us designate the torques transmitted by the friction clutches shown in Fig. 78*a*, *b* as M_{I} and M_{II}, respectively.

For the relations presented in Fig. 78*b*

$$\frac{M_{\text{II}}}{M_{\text{I}}} = \frac{1}{\sin \alpha/2} \cdot \frac{R}{R_1} = \frac{1}{\sin 20^\circ} \cdot 0.9 = 2.7$$

Thus, the torque, transmitted by the clutch, with the weight of the bronze pieces remaining the same, will be increased nearly 3 times as compared with the original design presented in Fig. 78*a*.

In the design displayed in Fig. 78*c* each piece is divided into three parts: one internal (trapezoidal) and two lateral (triangular). The centrifugal force of such a composite piece, acting on the cylindrical friction surface of the driven component is equal (provided the total mass of the pieces and radius ρ are the same) to the centrifugal force developing in the first design (Fig. 78*a*). At the same time, the internal trapezoidal element acts upon the lateral elements in a wedge-like manner, thus causing additional transverse forces $\frac{P_{c.f}}{\tan \alpha/2}$ ($P_{c.f}$— centrifugal force of the internal element) which are taken up by the cheeks of the driven component.

In this case there are three surfaces of friction: one cylindrical and two plane.

The additional torque

$$M_{add} = M_{\text{I}} \frac{G'}{G} \cdot \frac{1}{\tan \alpha/2} \cdot \frac{R_2}{R}$$

where G'/G = weight ratio between the trapezoidal internal elements and complete pieces;

R_2 = friction radius on the driven component cheeks

The relation between the total torque M_{III} transmitted by the clutch and the torque M_I of the initial design will be

$$\frac{M_{III}}{M_I} = 1 + \frac{G'}{G} \cdot \frac{1}{\tan \alpha/2} \cdot \frac{R_2}{R_1}$$

For the relations given in Fig. 78*c*

$$\frac{M_{III}}{M_I} = 1 + 0.6 \frac{1}{\tan 20^\circ} \cdot 0.9 = 2.5$$

Thus, the separation of insert pieces without increasing their weight will increase the transmitted torque 2.5 times as compared with the original design.

If the design is changed to two rows of trapezoidal-section insert pieces (Fig. 78*d*), then each of them will act upon two surfaces, namely: conical (cheek) and plane (central disk of the driven component). All in all there will be four frictional surfaces in the clutch. Provided the total weight of the insert pieces and distance ρ are the same, the torque transmitted by this clutch will exceed that of the original one in the ratio

$$\frac{M_{IV}}{M_I} = \left(\frac{1}{\sin \alpha/2} + \frac{1}{\tan \alpha/2} \right) \frac{R_3}{R_1}$$

For the relations given in Fig. 78*d*

$$\frac{M_{IV}}{M_I} = \left(\frac{1}{\sin 20^\circ} + \frac{1}{\tan 20^\circ} \right) 0.9 = 5.2$$

The increased torque in this case has been achieved mostly by halving the wedge angle of the insert pieces in comparison to the design shown in Fig. 78*b*. However, a similar result can be achieved also in the design of Fig. 78*b* by decreasing the insert piece wedge angle from $\alpha = 40^\circ$ to $\alpha = 20^\circ$, but the specific loads on the friction surfaces in this case will be twice those in Fig. 78*d*.

Thus, for the same weight of the driving elements the transmitted torque can be increased five times by imparting to the clutch a more rational design, or, conversely, for the same initial value of the torque the weight and overall dimensions of the clutch can be substantially reduced by the same design measures.

(*c*) *Effect of Power Schemes*

The weight of a unit to a large extent is dependent upon the power scheme, i.e., on how the main acting forces in the construction are taken up and balanced. The power arrangement is considered rational if the acting forces are balanced over a short section by means

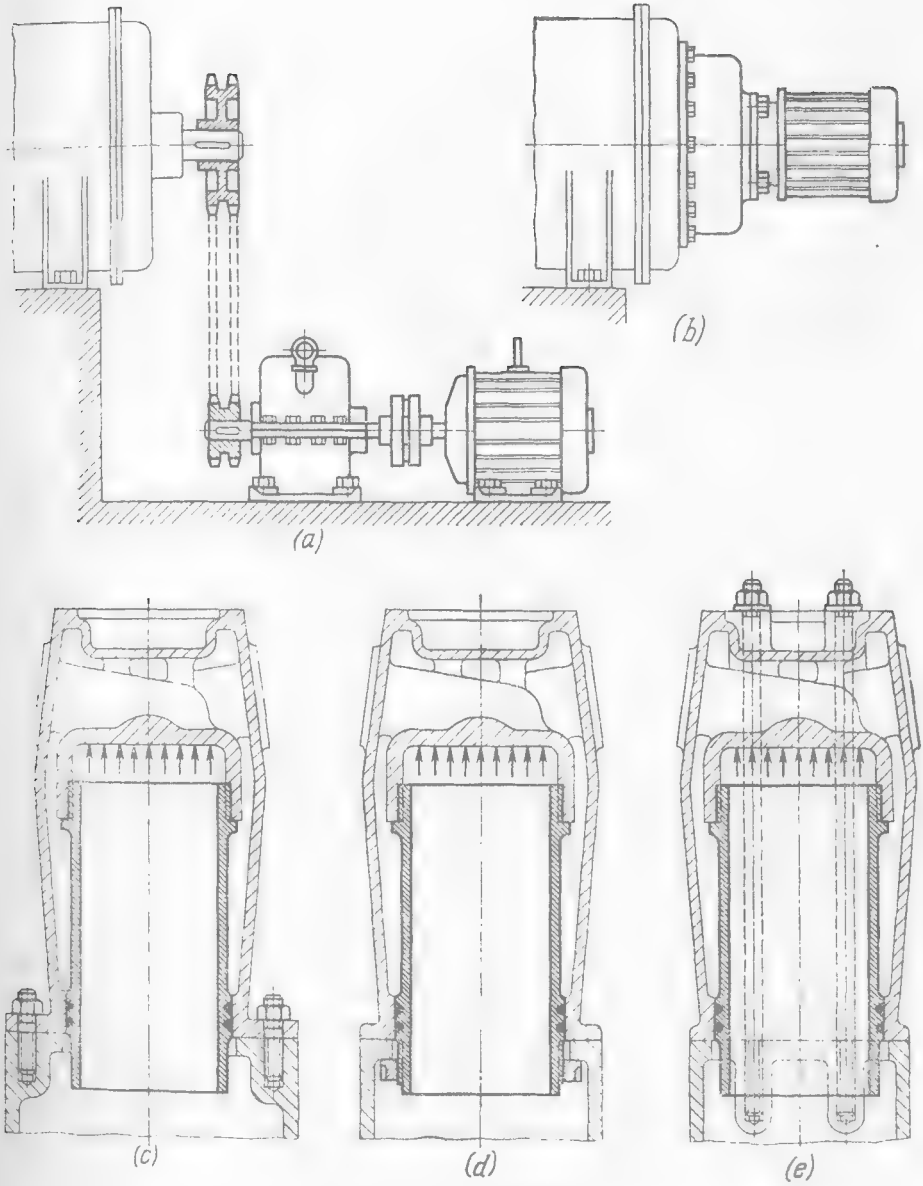


Fig. 79. Power schemes

(*a*), (*b*) **machine** drive; (*c-e*) cylinder blocks of internal combustion engines

of elements which operate preferably in tension or compression. To use the existing elements of a structure is better since the introduction of special elements is accompanied by an increase in the weight.

It is unreasonable to effect a machine drive by means of a chain drive from an electric motor through a reduction gear unit (Fig. 79*a*), as this causes transversely acting forces on the machine and reducer driving sprockets and produces additional loads upon the shafts and casings. The installation is distinguished by its large overall sizes. A more preferable drive is obtained from a flange-mounted motor through a co-axial reducer directly connected to the machine (Fig. 79*b*). In this case the reaction forces of the drive will be counterbalanced over the shortest path in the reducer casing, not bringing additional loads onto the system's elements. The overall dimensions of the entire installation are sharply decreased. In addition, all the driving mechanisms are enclosed, which facilitates lubrication.

Illustrated in Fig. 79*c-e* are typical schemes of power distribution applied in modern designs of internal combustion engines with detachable cylinder blocks.

Three ways of transferring the flash impacts to the crankcase are possible: through cylinder jackets (Fig. 79*c*), via cylinders (Fig. 79*d*) or through tie studs (Fig. 79*e*), named bearing jackets, bearing cylinders and bearing studs, respectively.

The disadvantage of the first system is that flash impacts are taken up by the cast jacket walls possessing lowered strength. Consequently, wall sections have to be thickened.

In Fig. 79*d*, the flash impacts are taking up by steel cylinder walls. The latter, due to manufacturing reasons, must have a certain minimum thickness. Therefore they generally have an ample safety margin, which allows them to withstand the action of gas pressure forces. Hence, they can be loaded by tightening without increasing their sections. This makes the bearing cylinder system principally the lightest one.

The version illustrated in Fig. 79*e* is heavier than the previous two because of the tie studs, the role of which in the first two cases were fulfilled by the existing elements: jacket (Fig. 79*c*) and cylinders (Fig. 79*d*).

Figure 80 shows power arrangements of turbine rotors fitted with a series of blades.

The one-piece forged massive rotor *1* is quite unsuitable because of its weight. The second design *2* is somewhat better as it has lightening recesses at the ends.

The drum-like hollow rotor *3* has a small weight, but its strength and rigidity are insufficient to withstand the action of the blade centrifugal forces.

In the versions *4*, *5* and *6* the drum-shaped rotor is strengthened internally against bow by ribs.

In the strongest and lightest designs *7* to *12* the blade centrifugal forces are taken up by disks working in tension.

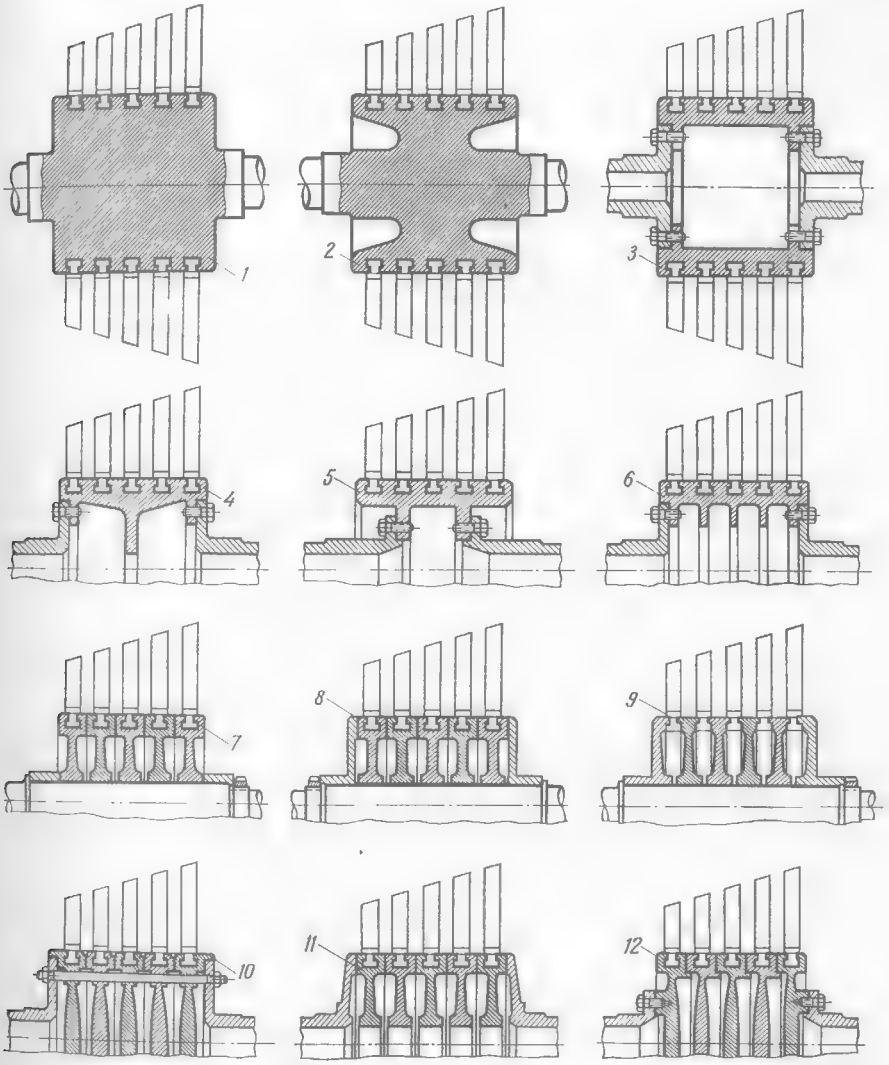


Fig. 80. Power schemes of turbine rotors with insert blades

The disks can be united into a single unit on a central shaft (rotors *7-9*), clamped by means of peripheral bolts (rotor *10*) or welded (rotors *11-12*).

In design *7* the disks are tightened on the central shaft against a hub, which causes undesirable bending stresses. This drawback is eliminated in design *8* where the disks are tightened at the rims.

In an original rotor design *9* the disks are positioned between the blades, this facilitating manufacture of slots and blade assembly.

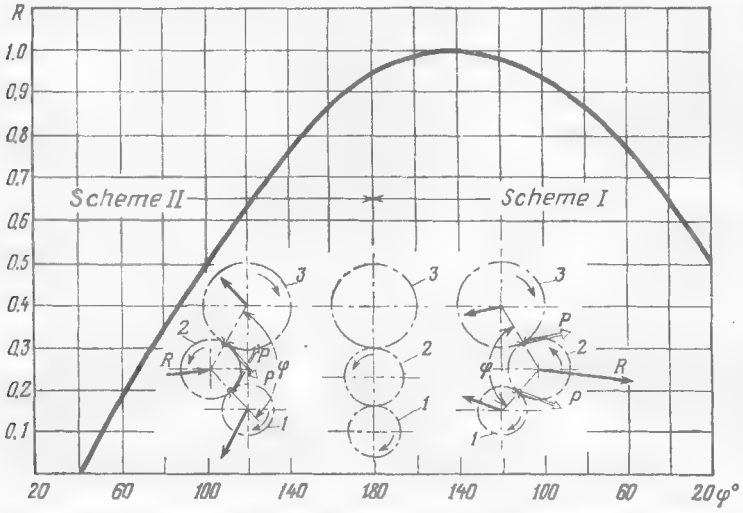


Fig. 81. Diagrams of gear drives with an intermediate gear

Welded constructions have two forms: in the first (rotor *11*) the disks during welding are centred one relative to another with the aid of a false-arbor inserted through centring disk bores which lower the disk strength.

In version *12* the disks are centred on spigot diameters near the rims. This enables the disks to be made as a continuous assembly.

An example of improved power distribution scheme in a gearing unit incorporating an intermediate gear is depicted in Fig. 81. The arrangement of the intermediate gear greatly affects the amount of load acting upon the gear supports.

Let the small gear *1* be the driving pinion running clockwise. It is not wise to position the intermediate gear to the right of the drive axis (scheme I), because the drive forces P acting upon intermediate gear *2* add vectorially and produce a substantial force R which loads the gear supports.

It is more advantageous to position the intermediate gear to the left (scheme II). In this case forces P when vectorially added will to a significant degree neutralize each other and the resultant force R loading the supports of gear *2* will be considerably less.

The magnitude of the resultant force R will depend, in both cases, upon angle φ between the axes *2-3* and *2-1* of the gear centres.

For scheme *I*

$$R = 2P \sin(\varphi/2 + \alpha) \tag{3.19}$$

For scheme *II*

$$R = 2P \sin(\varphi/2 - \alpha) \tag{3.20}$$

where P = driving pinion peripheral force;

α = pressure angle (in standard gears $\alpha = 20°$)

Figure 81 shows variation of force R as a function of angle φ. The value of force R for the most disadvantageous case, i.e., when the intermediate gear is on to the right and $\varphi = 140°$ [$R = 2P \sin (70° + 20°) = 2P$] is taken as a unit force.

From the graph, the magnitude of force R in scheme *II* with the same values of angle φ is much less than that in scheme I. For example, when $\varphi = 100°$ the resultant force in scheme *II* ($R = 0.5$) is nearly half that in scheme *I* ($R = 0.95$).

Hence, such a simple constructional measure as the repositioning of the intermediate gear from one side to the other can assure a significant advantage in the power arrangement and considerably lower the level of the acting forces.

When $\varphi = 40°$ in scheme *II* $R = 0$. Actually this is an unreal condition since the intermediate gear would require an enormous diameter. The least real values of angle φ are 80-60°.

However, should the rotational direction of pinion *1* be changed from that in the Figure it will be more profitable to position the intermediate gear to the right. The same will be true if, with the given direction of running, the driving gear is the large gear *3*.

The position of the intermediate gear may be decided by a general rule: the best position for the intermediate gear is on the side where the force of the driving gear seems to push it into engagement. The most disadvantageous position is when the driver tends to force the intermediate gear out of engagement.

(*d*) *Multi-Limb Power Schemes*

Considerable weight savings can be achieved by the application of multi-limb power schemes, i.e., by distributing the power through several parallel channels. The transmission of torque through the agency of several, parallel working gears (stage gearings, multi-satellite planetary gearings, etc.) is an example.

Figure 82 illustrates the overall dimensions of two different gear systems: a single-limb (Fig. 82*a*) and a four-satellite planetary mul-

ti-limb system (Fig. 82*b*) with identical gear ratios. A comparison of the overall dimensions clearly shows the advantage of the multi-limb version.

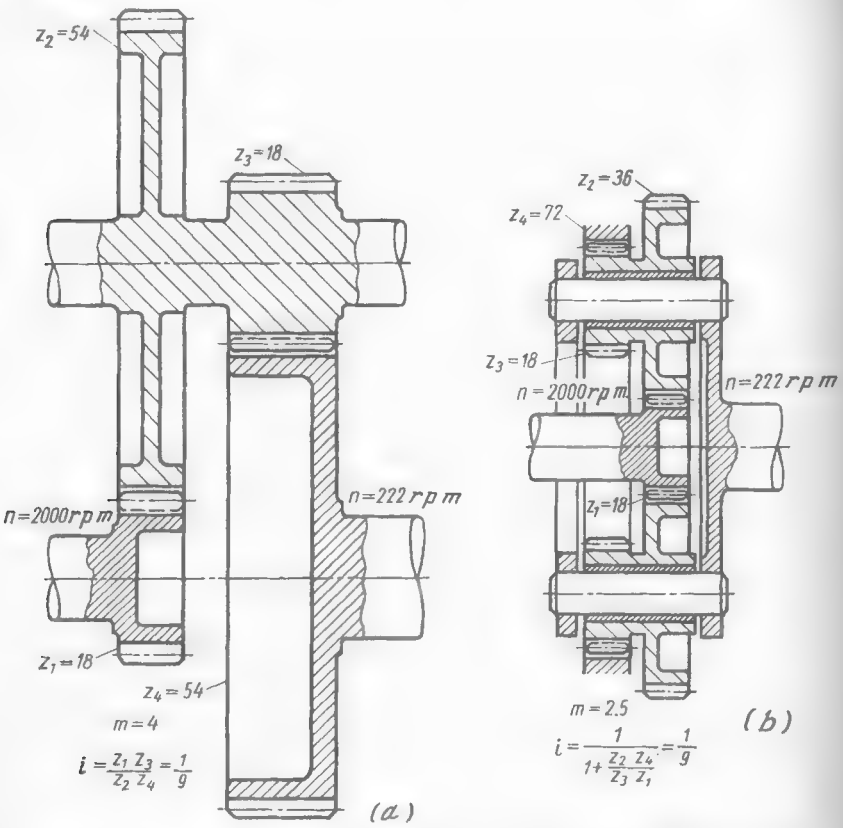


Fig. 82. Comparative sizes of (*a*) single- and (*b*) milti-limb transmissions

(*e*) *Rational Selection of Machine Parameters*

Significant savings may also be attained by a rational selection of machine parameters. As an example Fig. 83 shows two internal combustion engines of equal power with stroke-to-diameter ratios of $S/D = 1.5$ (Fig. 83*a*) and $S/D = 1$ (Fig. 83*b*).

The small height, typical of short-stroke machines can be reduced still more by reducing the height H of the piston (in Fig. 83*b*, $H = 0.75D$; in Fig. 83*a*, $H = D$), or by decreasing the ratio of the connecting rod length L to the crank radius R (in Fig. 83*b*, $L = 3.2R$; in Fig. 83*a*, $L = 3.8R$). Combination of these two measures enable the sizes and weight of the machine to be substantially reduced.

It is necessary to stipulate the increase in piston force carrying capacity because reducing piston height and the L/R ratio leads to a specific pressure increase on cylinder walls.

In some categories of fluid- or gas-operated machines (hydraulic presses, air- and steam-operated hammers, air- and hydraulically-operated drives) a significant reduction in size and weight can be accomplished by increasing working pressures of liquids (gas). Up to certain limits the same is applicable to internal combustion engines (supercharging and higher compression ratios), which allows the displacement to be decreased or, for the same displacement, the power to be raised.

Occasionally, for example, in some machine power generators, it is possible to reduce the weight by increasing the rotational speeds.

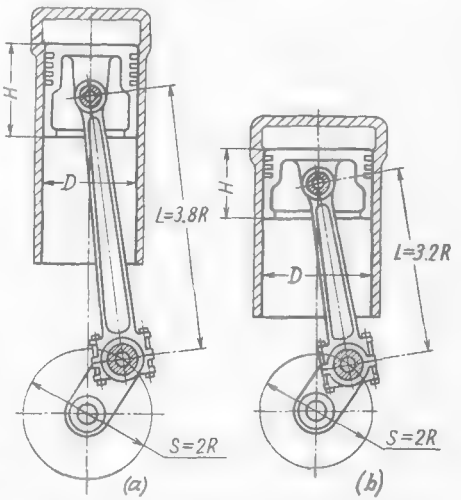


Fig. 83. Comparative sizes of (*a*) long- and (*b*) short-stroke piston engines

This method has its limitations. In internal combustion engines increasing the number of revolutions per minute is limited by gas suction speeds accompanied, in turn, by insufficient filling of cylinders and less litre power. In steam and gas turbines increased revolutions mean a corresponding rise of fluid through rates, causing a rise in internal losses.

Furthermore, higher speeds increase dynamic loads and wear.

It is also necessary to consider the characteristics of machine-consumers of energy. If the revolutions of a machine-energy consumer are stipulated, then, increasing the revolutions of a machine-generator will require the use of a reduction unit or a greater degree of reduction in the existing reducer which increases the weight of the installation.

Thus, careful consideration must be given to the positive and negative sides of the argument when higher rotational speeds are used as a means of saving weight.

High weight reductions can be obtained by the transfer to principally new designs and processes. Thus, steam engines were replaced by steam turbines as the latter have much higher power/weight ratios. High-power internal combustion piston engines are being ousted by gas turbines. Steam turbines apparently in time will give

way to gas turbines which do not require large auxiliaries (boilers, condensers). In the field of electrical energy installations the root of the change will be brought about by MHD generators which directly convert thermal energy into electrical energy.

3.4. Correction of Design Stresses

General calculation methods allow stress determinations to a satisfying degree of accuracy only for a comparatively small number of simple loading schemes. In several cases the magnitude and distribution of stresses inside the body of a component part cannot be calculated. The majority of casings, mounting frames, beds, crankcases belong to the incalculable category.

The effectiveness of the method of correcting design stresses and lowering strength reserves as a means of lowering the total machine weight depends on the weight correlation of the calculable and incalculable parts.

Returning to the calculable parts, it must be pointed out that calculations are based on simplifications not always holding true in real conditions.

The most important factors determining the deviations of the actual values of stresses and strength reserve from the calculated ones are as follows:

scatter of material strength characteristics (ultimate strength, yield and fatigue) from their rated values which are averaged statistically from a series of tested specimens;

heterogeneity of material; scatter of strength characteristics in different parts and different points of sections;

changes of materials strength as a function of the loading character (rate of loading);

deviation of the design loading scheme from actual loading conditions;

deviation of actual acting force magnitudes from their nominal values;

deviation of actual stresses from nominal stresses caused by the system's resilience;

disregard of strength and rigidity characteristics of parts adjacent to the part being calculated;

formation of local stresses at the component fixing points and the points of force application;

formation of additional forces and stresses, caused by inaccurate manufacture, assembly and installation (e.g., high edge pressure due to a misalignment or distortion of supports);

overloads in operation;

internal stresses originating during the manufacture owing to macro- and micro-heterogeneities of the material.

(*a*) *Local Stresses*

Machine elements are usually rather short and the loads are applied relatively close to each other.

Local stresses occurring at the spots where forces are applied, as well as in supports and fixing points, propagate into the depth of material, at times throughout the entire length of the part. Consequently, the actual magnitudes of stresses and character of their distribution differ strongly from the calculated values.

Another feature typical of engineering components is the complexity of their forms and change in sections. At section transition points stress concentrations occur.

Thus, in machine components a very large role is played by local stresses which at times are the deciding strength criteria.

For example: in the conrod of a piston engine (a compressor or internal combustion engine) its stresses formally determined by calculation from the action of gases and inertia forces have a value close to the actual stress value only at the section midway along the length of the conrod quite far from the big and small ends. The stresses prevailing in the conrod ends and in the points where the ends are mated with the rod, are very complicated, particularly, in the conrod split big end or when knuckle conrods are involved (V- and W-type machines). The type of stressed state, as well as the magnitude and distribution of stresses in the conrod end are influenced by many factors, in particular by the tightening forces and resilience of conrod bolts, configuration and rigidity of the end, and by the crankshaft rigidity where the crank neck joins the conrod.

To account for all these factors is difficult. True stress values are often only defined by analyzing fractures, cracks and breaks which appear after the conrod has run in the machine.

When a hollow cylindrical shaft supported at the ends is subjected to bending, a simplified calculation scheme is used assuming that the load is concentrated in the centre, halfway between the bearing surfaces (Fig. 84*a*), or that it is uniformly distributed throughout their length in the plane of acting forces (Fig. 84*b*), and the stresses are determined from a simply supported beam. The schemes do not consider the actual force distribution along the length and circumference of the supports or the influence of transverse load components on the strength and rigidity of the part, nor the influence of support rigidities on the load distribution or even the values of edge pressures and local stresses in the loaded areas.

If a part is interference mounted in its supports, then additional compressive and crushing stresses occur in the seating areas. If the mounting has a loose fit, shocks will accompany any alternating or pulsating loads, causing additional stresses.

A true loading picture can be imagined (with a sufficient degree of confidence) for a particular case if we assume that the part is mounted in plain liquid-friction bearings and that the transversely acting force is taken up by a plain bearing (e.g., a floating gudgeon pin of a crank and conrod assembly).

It is well known that the pressure distributed along the bearing oil layer changes in accord with a parabolic curve whose maximum

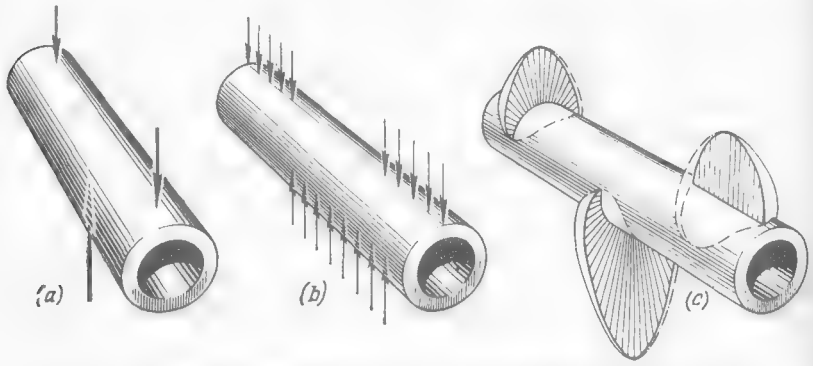


Fig. 84. Loading schemes for an axle supported at two points

ordinate is 2.5-3 times greater than the mean specific pressure $k\left(k = \frac{P}{ld}\right)$. In the transverse section the pressure is distributed over an arc of 90-120° (Fig. 84*c*).

From a comparison of formal loading schemes (see Fig. 84*a*, *b*) it becomes obvious that the first scheme overestimates the stresses occurring in the critical section of the part, and the second scheme underestimates these stresses; neither of these schemes considers the transversely acting components of the load and the stresses and strains produced by these components.

The load distribution picture over the oil bearing layer depicted in Fig. 84*c* is closer to the actual condition. However this picture may substantially change owing to the elastic deformations of the shaft and bearings, the emergence of increased edge pressures, etc.

The local stresses and strains, in the areas of force application, can reach significant values and decisively affect the serviceability of a part.

Let us consider a typical practical case.

The end of a crankshaft mounted in plain bearings was loaded with a comparatively modest load P from a gear positioned between the supports. Usual strength calculations for the crankshaft with permitted stress values of 20 kgf/mm^2 led to the shaft end configuration shown in Fig. 85*a*. The recurring front bearing failures made the

designers examine the unit more carefully. When the crankshaft was bench-tested and subjected to an acting force equal to the designed one, it was proved that the shaft end deformed, acquiring an elliptical cross section whose major axis exceeded the bearing diameter by 0.2 mm. Meanwhile, with a diametral clearance of 0.1 mm an ellipticity amount of only 0.05 mm eliminates completely the wedge shape of the oil layer in the zone of the maximum closeness between the shaft and bearing, which is indispensible for the correct functioning of the bearing.

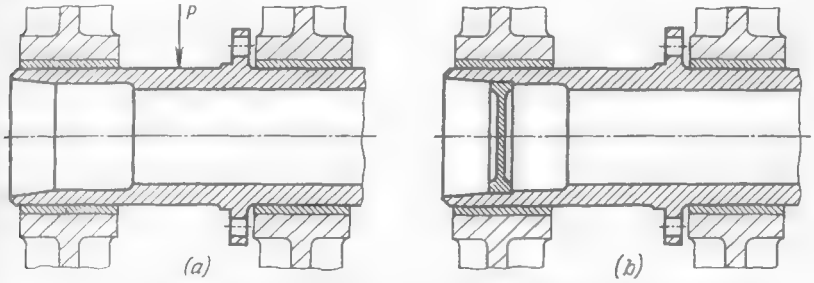


Fig. 85. Increasing the rigidity of a crankshaft by pressed-in plug

This defect was rectified very simply without serious weight penalty. A plug press-fitted into the shaft end (Fig. 85*b*) greatly increased the rigidity of the part and enabled the true cylindricity of the shaft to be kept.

The arbitrariness of strength calculations can be illustrated by an example of pure shear. With usual stress calculations it is generally assumed that a part operates over its full section and that shearing stresses are the same throughout the entire section and equal to the quotient obtained from the division of shear loads by the area of the section.

However, the actual picture of the stress distribution sharply differs from the described scheme. The mechanism of shear has been well studied on the punching of sheet materials. Along its contour the sheared section is subjected to high crushing and shearing stresses under the action of which in the material there occur, at first not deep, microflaws and plastic shear before the bulk of the material begins to respond.

With reference to common engineering parts this means that the serviceability of parts is disturbed long before the shear stresses in the section attain dangerous values. In fact, the part fails as a result of stress concentrations in the surface layer, accompanied by local crushing and plastic deformation at the place where the shear force is applied. This phenomenon is sharply expressed in the shear of cylindrical parts, when the stresses are concentrated over a small arc

of the surface close to the acting force. The softer the material of the part being sheared in comparison to the punch and the higher the punch rigidity, the greater the crushing.

After crushing the surface layer of a part becomes stronger due to work hardening and the bulk of the metal entering the action. The joint, however, is proved spoilt due, in the first instance, to the sharp drop in strength from local cracks and flaws which become stress concentrators with successive load applications and, secondly, to the disturbed joint geometry caused by the onset of crushing.

From the above discussion we may derive the following important rule for designing connections working in shear: the hardness of the cutting material and the material being cut must be as equal as possible; the greater the surface hardness of the parts, the more reliable the insurance of the connections against fracture.

The complexity of shear stress distributions cannot be accounted for by elementary calculations. Such an important factor as the surface hardness of a material, which determines the connection's serviceability, is also neglected.

(*b*) *Effect of the Resilience of System*

Formal calculations do not usually account for resilience characteristics of a constructional system, which in practice have a great effect on the true stress values.

As an example we consider a shaft, a component widely used in machine building, supported by its ends and subjected to a bending load in the centre from a connecting rod. For the sake of simplification we neglect the influence of transverse load components, as well as support reactions, keeping during the calculations to the standard schemes of load distribution along the axis of the part.

If the rigid sections are assumed to be at the connecting rod centre and at the edges of the supports (scheme 1, Table 6), then it is possible to consider that the shaft is acted upon by concentrated force P at the centre and that the bearing reactions are applied at the extreme shaft points separated by span l.

With this loading scheme the stresses in the critical section in the shaft are

$$\sigma_1 = \frac{Pl}{4W}$$

where W is the resisting moment of the shaft section.

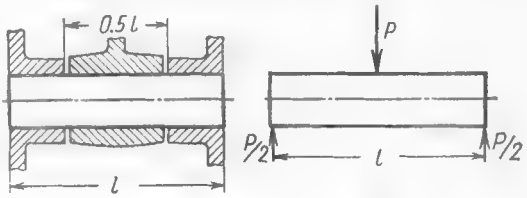
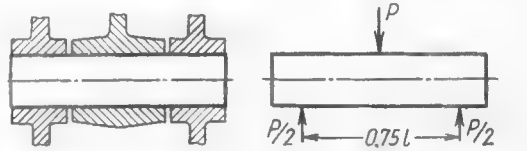
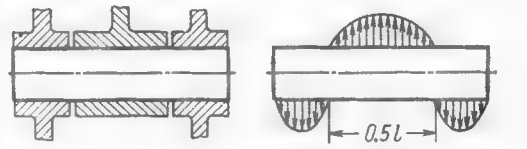
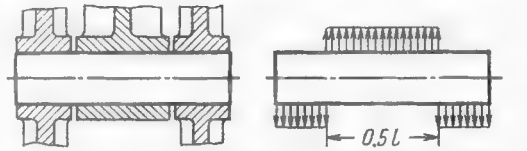
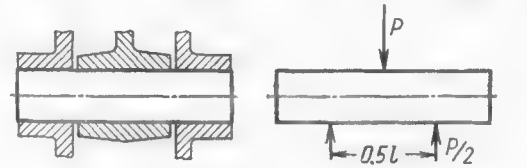
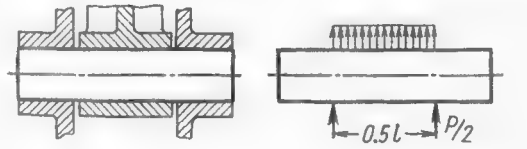
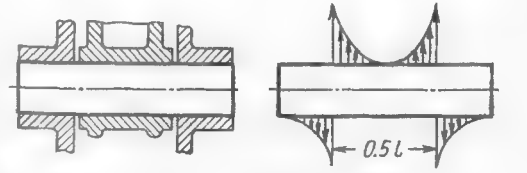
The maximum flexure of the shaft

$$f_1 = \frac{Pl^3}{48EI}$$

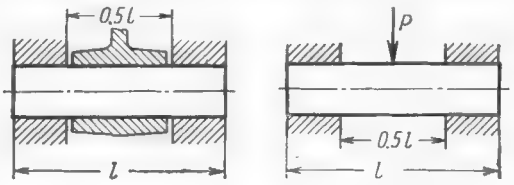
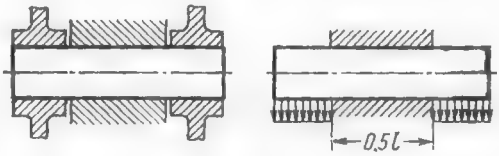
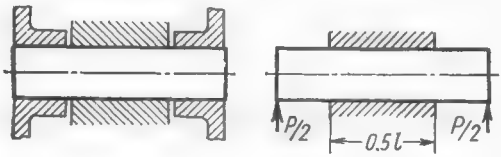
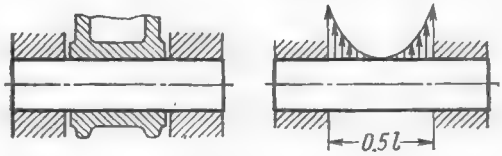
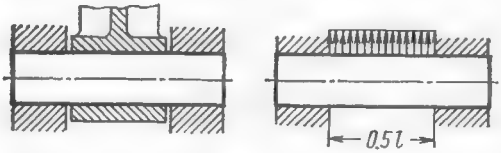
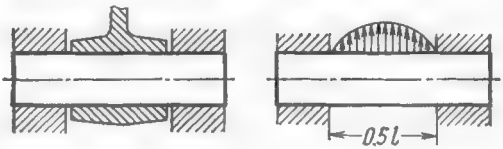
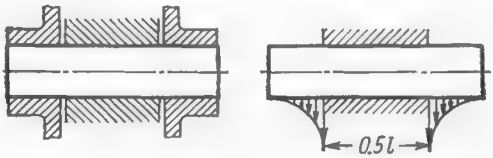
where I = moment of inertia of the shaft section;
E = Young's modulus for the shaft material

Table 6

Load Distribution Schemes

Scheme No.	Load distribution sketch	σ	f
1		1	1
2		0.75	0.42
3		0.56	0.37
4		0.5	0.27
5		0.5	0.125
6		0.25	0.077
7		0.25	0.015

Continued

Scheme No.	Load distribution sketch	σ	*f*
8		0.25	0.031
9		0.2	0.021
10		0.167	0.0157
11		0.04	0.0035
12		0.5	0.125
13		0.25	0.077
14		0.08	0.012

Assuming these values of σ and f to be equal to unity, we consider how the strength and rigidity of the system are changed with the loading schemes.

In the designs where the rigid sections of the supports are at their centre it is possible to assume the following probable schemes of the acting forces: bending by the point load P with a span equal to $0.75l$ (scheme 2), and bending by a load distributed parabolically (scheme 3).

Under such circumstances we shall obtain respectively

$$\sigma_2 = 0.75\sigma_1; \quad f_2 = 0.42 f_1$$

and

$$\sigma_3 = 0.56\sigma_1; \quad f_3 = 0.37 f_1$$

The uniformity of the load distribution is improved with an increase in the rigidity of the connecting rod and supports (scheme 4).

In this case

$$\sigma_4 = 0.5\sigma_1; \quad f_4 = 0.27 f_1$$

The strength and rigidity of the design can further be enhanced by transferring the rigid sections to the supports inner ends (schemes 5, 6 and 7). Depending on the connecting rod rigidity and on the assumed scheme of load distribution, we obtain the following stress and flexure values

$$\sigma_{5\text{-}7} = (0.5 \text{ to } 0.25)\,\sigma_1$$

$$f_{5\text{-}7} = (0.125 \text{ to } 0.015)\,f_1$$

The strength and rigidity will grow still more (schemes 8 to 11) with encastre ends (without clearance or with an interference). Depending on the connecting rod rigidity and the law of load distribution, the values of σ and f will vary within the following limits:

$$\sigma_{8\text{-}11} = (0.25 \text{ to } 0.04)\,\sigma_1$$

$$f_{8\text{-}11} = (0.031 \text{ to } 0.0035)\,f_1$$

With fixed mounting in the rod (also without any clearance or with an interference), when the shaft ends are cantilevered (illustrated in schemes 12, 13, and 14) the gains in strength and rigidity are less. This is attributable to the inferior strength and rigidity characteristics inherent in cantilevered systems in general. In this case the magnitudes of σ and f are

$$\sigma_{12\text{-}14} = (0.5 \text{ to } 0.08)\,\sigma_1$$

$$f_{12\text{-}14} = (0.125 \text{ to } 0.012)\,f_1$$

The general conclusion, that can be drawn from the above review, says that the resilience of a system and the manner in which loads are distributed have immense influence upon strength and rigidity.

Within the range of the above described schemes, the stress value can be 25 times and flexure, approximately 300 times (scheme 11) less than those in the original scheme 1.

The gain due to the system's resilience can be realized by making the design as rational in shape as possible. However, it should be noted that any assessment of the system characteristics, particularly the law of load distribution along the axis of a part, is unavoidably somewhat arbitrary. Hence, the above-listed relationships are rather

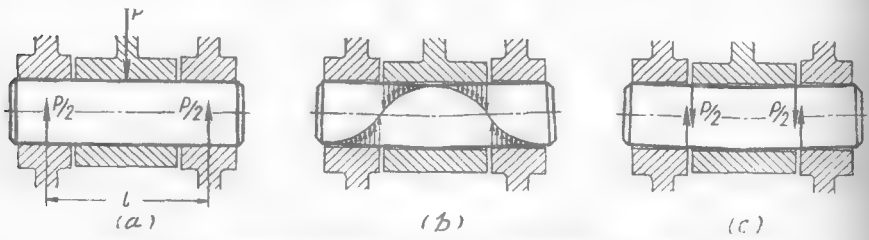


Fig. 86. Loading schemes for a pin supported at two points

construction recommendations. Their value as to the calculation accuracy is relative because they only indicate probabilities for the given form of load distribution.

It should also be noted that the loading scheme and law of the load distribution are dependent not only upon the design of the unit but also on its deformability which is determined by the level of the acting stresses, the material of the part (Young's modulus), etc. With the given design of the unit the loading scheme sets in on its own accord as a result of the interaction of the loads and deformation developing in the unit.

We can explain this by the same example of bending a simply supported shaft which has its fixed section at the centre between the supports (Fig. 86). The loading scheme illustrated in Fig. 86*a* is possible with small loads or with a high rigidity of the system. As the force grows (or section rigidity lowers) the system is deformed, as shown in somewhat exaggerated manner in Fig. 86*b* (for the sake of simplicity only the shaft deformation is shown). The deformation acts as a strengthening agent causing load concentration on the edges of the support surfaces. As a result a new system of acting forces arises, which follows a triangular law or (as shown in the Figure) a parabolic one. In this case the stresses are sharply lowered, and still more sharply decreases the deformation.

As the loads increase (or the section rigidity falls) the scheme approaches that of pure shear (Fig. 86*c*), which lowers stresses and deformation even more.

Thus, the load increase is accompanied by a self-strengthening process caused by the development of deformation which contributes

to a more favourable load distribution. However, the deformation simultaneously increases the system rigidity acting reversely and at a certain stage equilibrium is reached, fixing the load distribution and defining the true values of stresses and strains occurring under the loads. The actual instantaneous strength and rigidity of the system are fully dependent upon the magnitudes of the loads and rigidity of the areas which transfer and take up the loads. In most cases

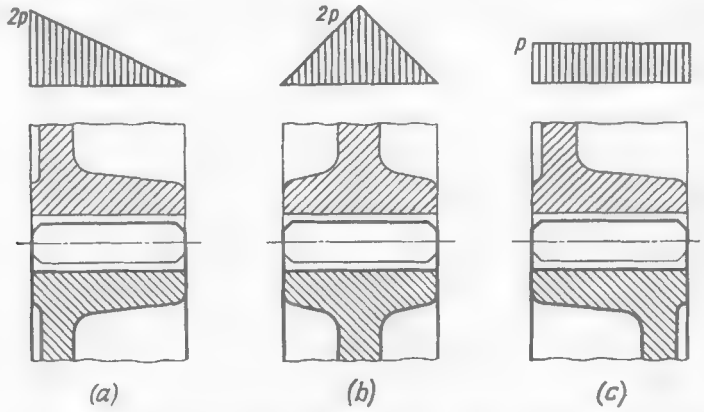


Fig. 87. Load distribution along a gear tooth for differently designed gears

the fashion of loading cannot be established by calculation. Thus it is clear that a system, self-adapting to load conditions, arrives at a state intermediate between the extremes depicted in Fig. 86*a*, *c*.

Let us consider one more example of the effect of rigidity on stress values: the load distribution over the tooth length of two gears in mesh (Fig. 87). The character of load distribution and its maximum value depend on the relative position of gear webs. If the webs are located in the same plane close to the end face of the gear (Fig. 87*a*), then the load is concentrated mostly at the rigidity node, i.e., in the plane of the webs. The remaining parts of teeth positioned on a relatively elastic rim are loaded less. The probable load distribution in this case is shown by the triangle, with its apex found in the plane of the webs. Maximum load on a unit length tooth equals $2p$, where p—average load, with the usual assumption that the load is uniformly distributed over the entire tooth length.

If the webs are positioned in the plane of engagement symmetry (Fig. 87*b*), then the picture of load distribution is expressed by the triangle having its apex in the plane of symmetry. Maximum load as in the previous case is equal also to $2p$.

The load on the teeth will be distributed more evenly when the webs are arranged on different sides relative to the plane of engagement symmetry (Fig. 87*c*).

(c) *Effect of the Strength of Mating Parts*

In practical calculations the strength of parts (hubs, bushes, supports, etc.) mating with the designed part is often neglected. The part being designed is generally treated in isolation and the effects of adjacent parts, which transmit or accept the load, are considered (and not always) in the design scheme of the load distribution along the part. This is admissible only when the length of a mated piece is small in comparison with the length of a part being designed, or if these two parts are mated with clearance. If the length of the mated part is commensurable with that of the part being designed (especially when the fit is without clearance or with interference) any disregard of the mating part can result in quite sizable errors.

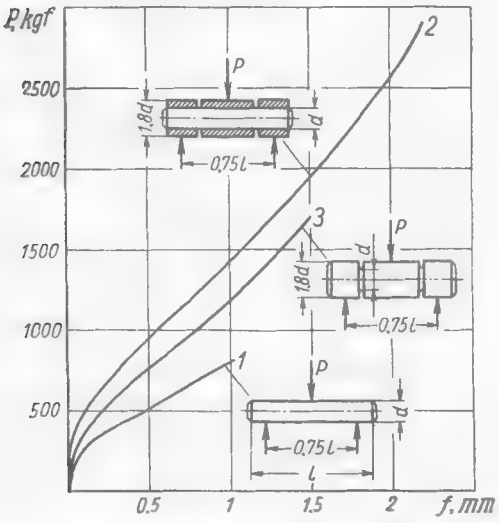


Fig. 88. Effect of conjugated parts upon strength and rigidity

Let us again consider the same case of a simply supported axle loaded with transverse force *P*. In Fig. 88 (in a force-deflection coordinate system) are shown the results of testing three specimens made from steel grade У8А heat-treated to a hardness of 45 Rc. Specimen *1* is a rod 10 min diameter and 80 mm long. Specimen *2* is a similar rod upon which three bushes are slide-fitted, each bush being 18 mm in diameter. The two bushes at the rod ends imitate support bushes, while the middle one imitates the bush of a conrod. Specimen *3* (a reference specimen) is a rod 18 mm in diameter with two annular grooves corresponding to the end clearances between the bushes in specimen *2*.

The specimens were tested to failure. Specimen *1* failed at an 800-kgf load and a deflection of 1 mm; specimen *2*, at 2900 kgf and 2.2 mm. The composite specimen *2* proved stronger than the large solid specimen *3* having the same configuration, which failed at a load of 1700 kgf and 1.5-mm deflection. This can be explained by stress concentrations at the undercuts.

With similar strains the composite specimen proved to be approximately twice as strong, and with similar loads, 3-5 times more rigid

than the plain one and as to the failure load, 3.7 times stronger.

The obtained values refer to the region of plastic deformations. Nevertheless they illustrate the fact that calculations which disregard the effect of mating parts are only very approximate.

(*d*) *Deviation of Acting Forces from Nominal Values*

Another reason for calculation inaccuracy is the difficulty of defining in a number of instances the actual values of the acting loads. This is particularly true of changing, pulsating and impact loads.

Let us take for example, a well investigated mechanism, a crank-and-connecting-rod assembly. In internal combustion engines the initial data for the strength calculations is the maximum gas pressure forces acting on the piston. It would seem that in determining of these forces there can be no mistake. However in practice the value of these forces and the stresses caused by them in the links of the mechanism depend upon many factors, but most of all, on the resilience and mass of the links.

Some of the flash energy is spent on the work of the resilient extension of cylinder walls, fixing studs and crankcase (within the limits of elastic strains). Another part of the energy is consumed in compressive deformations of the piston and connecting rod and also in the bending of the gudgeon pin, bending and torsion of the crankshaft, and expulsion of the oil layer from the clearances between the mating parts. A substantial portion of energy is spent on acceleration of reciprocating and rotating members. The largest part of the energy is returned and recovered at later stages in the cycle but energy is wasted on work of viscous shear, expulsion of oil layers from clearance gaps and hysteresis during elastic strain.

The greater the resilience of the system (i.e., the longer are the fastening studs, the more yielding is the crankshaft, the lesser are the cross sections and inertia moments of the parts, as well as the modulus of elasticity of the material), the smaller is the actual force stressing the parts and the weaker the forces reaching the final links of the mechanism. The introduction of resilient connections into the system, e.g., provision of spring couplings between the shaft and the final member (flywheel, screw propeller, generator, reducer), elastic torsion-bar suspension of the motor, etc., will sharply decrease maximum stresses in the system.

Any increase in the mass of intermediate links increases the instantaneous value of maximum forces acting upon preceding links, and lessens the forces acting on the subsequent links. The action of the enlarged mass of intermediate links is similar to that of an anvil, which absorbs the energy of an impact and protects other parts from higher stresses.

Of certain importance is the rate at which the pressure of operating gases is built up at the moment of ignition. The greater this rate, i.e., the closer the load approximates an impact one, the greater are the stresses in the system. However, the strength of the material increases with the increase in the loading rate.

(*e*) *Internal Stresses*

Internal stresses, which are inherent in any material, arise during manufacture and operation. The actual strength of a part depends on the interaction between internal stresses and stresses caused by external loads.

When establishing admissible stress values the previous history of the part is often neglected, as well as its future history, i.e., the gradual changes of mechanical properties of the part material during machine operation. Such changes can both weaken and strengthen the material. The weakening factors include corrosion, wear and damage of part surfaces; accumulation of microdamages as a result of multiple-repetitive loads; local tempering, which follows the heating brought about by cyclic loads.

The strengthening factors include the so-called material "training" processes caused by short-time stresses which exceed the yield limit; deformation strengthening caused by the structural changes in the stressed microvolumes of the material; spontaneously developing ageing processes accompanied by recrystallization and spread of internal stresses. Positive effects are felt from the design's adaptability, i.e., total or local plastic deformations, arising under the effect of overloads and causing load redistribution. A definite strengthening effect is obtained at the first wear stage (smoothing of surface microirregularities), favouring an increase in actual contacting surface areas, lowering the pressure peaks and levelling out the load on the surfaces.

It is generally accepted that defects appearing during the manufacture and use of a part are mostly random ones. This circumstance partly explained by the dispersion in the strengths characteristics of parts, a fact which is well known. Amongst components in one and the same batch, all of which were subjected to similar processing, some have high durability, while others will fail within quite a short period of use as a result of unnoticed initial defects or defects occurring in use.

Internal stresses are generally divided into the following three categories:

the first—those developing in a substantial volume of the part due to the heterogeneity of the metal's macrostructure (sometimes arbitrarily called macrostresses);

the second—those developing in the volume of the crystallites or in a group of crystallites due to the heterogeneity of the crystallite metal structure (sometimes arbitrarily called microstresses);

the third—those developing in the submicroscopic volumes due to defects and distortions in the atomic-crystalline lattice (sometimes arbitrarily called submicrostresses).

Stresses of the first group are often the result of technological processes which the component undergoes during forming operations. Since such technological treatment is a many-stage process, the stresses existing in the finished component are the result of accumulated and interacting stresses arising at each stage of the process. Heterogeneities of an ingot pass to the forging (or rolled stock); hot-pressure processing introduces new heterogeneities. Machining operations although eliminating the nonuniformity present in the cut-off metal, causes redistribution of the stresses which had been introduced at the earlier stages, leading to new additional stresses at surface layers. Heat treatment partially eliminates stresses, from the earlier stages, but at the same time it creates stresses of its own.

The internal stresses appearing in castings are generally the result of a non-uniform crystallization in foundry and shrinkage of material when cooling. As a rule, the stresses are concentrated around extrusion defects, shrink voids, pores, etc., often obtaining large values and inviting fractures, local cracks and general fissuredness of the casting. Other defects, often found in castings, are burnt spots, entrapped slag, spalled particles of the mould, oxide inclusions, sulfides and silicides, local liquation and local dendricity.

The main source of internal stresses when hot working by pressure is the uneven metal flow through sections oriented differently to the action of the deforming tool. Especially often irregularities occur at points of discontinuity, at conjugation zones of different section thickness, at exterior and interior corners. Other defects include shearing of material, folding, spalling, overlaps, hair cracks.

Lemon spots or flakes as they are usually termed in the parlance of iron and steel manufacture (which are in fact non-welded hydrogen bubbles) are rather common defects in alloy steel.

High residual stresses occur during heat treatment, particularly when hardening is followed by rapid quenching. Because of the non-uniform removal of heat from surfaces and inner layers of metal, as well as from changing sections, zones of high stresses are formed, sometimes leading to quenching cracks. In materials with low hardenability the phenomenon is aggravated by the interaction between hardened and soft zones. The martensite zones which have the largest specific volume are subjected to compression by the action of the adjacent denser layers of troostite, sorbite or perlite structures in which reactive tension stresses arise.

Substantial stresses occur in surface layers in the process of machining. Plastic shear and destruction of metal in the cutting process are accompanied by residual tensile stresses arising in neighbouring layers. The rougher the machining (i.e., the thicker the cut-off layer and the greater the applied cutting force), the higher the residual stresses (e.g., in rough-turning steel residual tensile stresses reach 80-100 kgf/mm^2). The mechanical stresses are added to thermal stresses appearing in the zone of cutting as a result of heat liberation, as well as to stresses forming from the structural and phase transformations in the zones of intensive heat liberation.

Even fine machine operations, such as, for example, grinding, are accompanied by residual tensile stresses reaching 20-40 kgf/mm^2. The greatest particular defect in grinding is a burnt spot, which causes local tempering in chilled steels and the development of soft spots of troostite or sorbite. Conversely, normalized or structurally improved steels, when subjected to a high temperature and emulsion cooling, can harden and form hard spots of martensite.

High local stresses can also originate during welding from the local heating up to a melt-point temperature and subsequent cooling accompanied by material elongation in the weld seam. Local stresses also arise in the zones of weld defects (incomplete penetration, poor fusion, loose areas, oxide and slag inclusions, etc.).

Stresses of the second group arise in the main because of the metal crystalline structure heterogeneity and due to differencies in physical and mechanical properties of the alloys phases and structures. Phases (e.g., ferrite, austenite, cementite and graphite in ferrous metals) have different crystal lattices, thereby their density, strength and resilience, as well as thermal conductivity, thermal capacity and thermal expansion characteristics are also different. Structures appearing as phase mixtures (e.g., perlite in steels) or hardened structures possess, in turn, such properties which make them different from neighbouring structures. Variations in crystalline orientation of metal grains cause anisotropy of the physical and mechanical properties in the microvolumes of metal. As a result of the combined action of these factors both intra- and inter-granular stresses come into existence, starting from original crystallization and following subsequent transformations in the process of cooling. At high temperatures the stresses are counterbalanced by the plasticity of material. However, they develop in low-temperature areas as a result of phase recrystallization and drop out of secondary phases (phase cold-hardening); with every total or local rise of temperature (due to the difference between heat conductivity and coefficient of linear expansion of structural components); due to the application of external loads (because of difference and anisotropy of mechanical properties); and also as a result of cold hardening, which follows total or local transition of stresses beyond the material yield limits.

Amidst other sources of the second group stresses are intra- and inter-granular foreign inclusions, microporosity, segregates, residual austenite (in chilled steels).

Stresses of **the third group** are due to numerous submicrodefects (dislocations) inherent in atomic-crystalline metal lattices. The fields of elastic stresses, forming around dislocations, can break the interatomic bonds, i.e., cause plastic deformation.

The third group includes also stresses which develop at the phase boundaries of phases with different crystal lattices (e.g., cementite and ferrite in Fe — C alloys, cuprous, magnesian and ferrous phases in Al alloys). Belonging to the same group are also the stresses occurring at the boundaries of subgrains (crystalline blocks) as a result of reorientation of the latter either due to heat treatment, or under the action of external loads, or else caused by cold hardening.

Occasionally submicrostresses can encompass extensive regions and turn into microstresses (for example, this is the case, when such stresses develop at grains boundaries owing to the difference between crystal lattices of grains and interlayers of the material).

Multiple distortions of crystal lattices, developing throughout extensive areas of material, can cause macrostresses, which will encompass a number of layers or the entire thickness of material (such are, for instance, the stresses occurring in macrovolumes due to the total plastic deformation of metal).

Numerous microstresses eventually turn into macrostresses covering extensive areas of metal or the whole metal volume. This can be illustrated by phase cold hardening, but it should be emphasized that phase cold hardening will lead to a higher dislocation density, as well as to distortions of crystal lattices and crystal blocks boundaries, thus producing submicrostresses throughout the entire metal volume.

It is obvious that classification of internal stresses into the three groups is rather arbitrary since all of these stresses are intermixed and may be of a local, regional or general nature.

Of significance from the practical point of view is that internal stresses can act either as weakening or as strengthening factors. One must bear in mind that internal stresses, whose sign coincides with that of the working stresses, are hazardous (e.g., tension-tension pair of stresses). Conversely, should the sign of internal stresses be opposite to that of the working stresses, the situation is favourable (e.g., tension-compression pair of stresses). Furthermore, it is noteworthy that internal stresses of the same sign are always accompanied by counterbalancing stresses of opposite sign in the neighbouring volumes. The relative value of oppositely-signed stresses depends on the extent of the material volumes in which these stresses are predominant. Thus, we may conclude that the critical factors deter-

mining the strength parameter are: first, the *position* and *orientation* of the stressed volumes relative to the acting working stresses; and, second, the *magnitude* of the internal stresses whose sign and direction coincide with those of the working stresses. The heterogeneities which produce areas of high tensile stresses, break the continuity of metal, and cause flaws and local plastic shear are, in effect, metal *defects*. The heterogeneities, which produce extensive zones of compressive stresses, contribute to metal consolidation and prevent the inception and propagation of plastic shear are, in fact, *strengthening factors*. They can be utilized to enhance the resistance of material to working loads.

Out of the weakening factors the most dangerous are macrodefects, as they create regions in which tensile stresses of the first group occur. Moreover, when working loads are applied, the macrodefects behave as stress concentrators, thus increasing still more the high level stresses.

Modern technology possesses efficient means of preventing and rectifying macrodefects.

Thus, the defects forming at the primary stage, during melting, to a significant degree are eliminated by melting in vacuum, in electric or electron-beam furnaces, by refining of steel, by electroslag remelting, etc. Ingot defects are lessened by vacuum casting, by assuring uniform crystallization of ingots, and also by using continuous-casting techniques.

Casting defects are eliminated by imparting rational shapes to castings, which assure more uniform crystallization and paralyze shrinkage effects; by properly chosen sand; by vacuum and die casting, etc.

Many types of microstresses are successfully eliminated by stabilizing heat-treatment. Crystal-structural defects of blanks, produced by hot plastic working, are eliminated by recrystallization annealing. Internal stresses in castings can be removed by low-temperature annealing (ageing).

Hardening stresses are reduced by special processes (step-by-step hardening, isothermal hardening, etc.). The stresses liable to appear because of limited hardenability are eradicated by the introduction of alloying elements (nickel, chrome, molybdenum and, particularly, boron).

The stresses, being caused by machining, are eliminated by proper selection of cutting conditions and removal of damaged layers by finishing operations (fine grinding, honing, super-finishing, power polishing, etc.).

In general the main task is to choose the most optimum production technique and observe the established production processes.

Macrodefects stealing in despite all the precautions should be discovered through careful inspection of all blanks at all stages of

their production by the application of high-sensitive methods (X-ray, magnetic and ultrasonic defectoscopy).

Deep crystal-structural defects are detected only by cutting micro-sections, i.e., through destruction of a part. In practice this method is applied on the basis of random sample testing.

The stresses of the second and particularly of the third group are, in effect, unavoidable. Here, the primary aim is not to eradicate the stresses (which would be impracticable), but rather to control their behaviour and utilize them for strengthening the material. Thus is the subject of the so-called *strengthening technique* which has extreme practical value.

(*f*) *Experimental Determination of Stresses*

Limitations and drawbacks of calculations compel the designer to determine stress values through experiments.

For the investigation of stresses the *optical-polarization* technique is of value. The method is based on the ability of certain transparent elastic materials to change their optical properties when acted upon by stresses.

The stresses in flat specimens are easier to determine. A specimen made of an optically-active material (generally organic glass), is placed in a beam of monochromatic polarized light and viewed through a second polarizer, whose light path intersects that of the first polarizer. Should the specimen be free of stresses, the second polarizer will extinguish the light beams that have passed through the first polarizer, and the specimen appears dark.

However, when subjected to a load, the specimen material will acquire a quality to divert the plane of polarization through an angle which is proportional to the magnitude of existing stresses. A part of polarized light passes through the specimen, which, owing to the interference of light beams, will be seen as streaks of alternate light and dark bands. The intensity and position of these bands will characterize the magnitude and direction of the existing stresses. When exposed to a beam of white light, the specimen will show coloured bands with continuously changing colours.

Such a method is applied when examining the distribution of stresses in regions with concentrated loads or in areas with stress concentrators (local weaknesses, sharp transitions, etc.).

Employment of this method for determining spatial distribution of stresses is very complex, which limits its value.

In recent years for experimental researches of surface stresses wide use has been made of *strain gauges.*

A strain gauge is a wire element which registers change of distance between two points on the specimen when the latter is affected by some load. The magnitude of the stress is determined indirectly from

the deformation on the basis of Hooke's law. Of the numerous types of strain gauges those most convenient and mostly used in practice are electrical resistance ones made as loops of wire or foil (Fig. 89*a*, *b*) 0.01 to 0.03 mm thick, glued to a strip of tough paper. The strain gauge is then glued upon the surface of the part to be tested in such a way that the length of its loops coincides with the direction of the expected deformation.

The measuring device consists of a current source, a galvanometer and a bridge comprising four equibalanced resistors, one of which is the strain gauge (Fig. 89*c*).

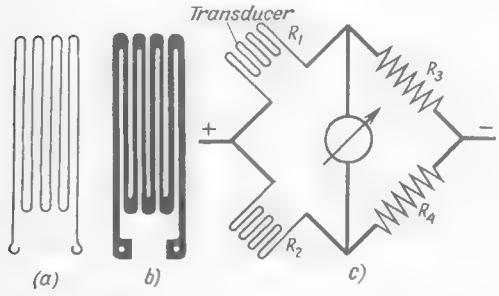


Fig. 89. Electric tensometers

(*a*) wire strain gauge; (*b*) foil strain gauge; (c) measuring bridge

The equation of balance is

$$\frac{R_1}{R_2} = \frac{R_3}{R_4}$$

Usually the R_2 resistor is also a sensor similar to R_1, and the R_3 and R_4 resistors are equalized, and the galvanometer set to zero. Any length change in the specimen changes the length of the strain gauge wire, thus altering its electrical resistance and causing a current, which is proportional to the strain, to flow through the galvanometer circuit. To exclude the effects of temperature the gauges should be made of elinvar or constantan.

If the deformation direction is unknown, it is advisable to position a number of strain gauges at 45° to one another. This will allow both the direction and value of deformation to be determined.

The base length of a strain gauge (i.e., the length of loops) is generally 20 mm. Yet, strain gauges with 3-5-mm bases are made to enable differential investigations of strains on intricately shaped components over tiny areas of surface to be carried out.

When studying quickly changing deformations (e.g., in the event of cyclic loads), an amplifier is introduced into the circuit and the deformations recorded by means of an oscillograph. This method is applied when testing pulsations. The specimen is installed on a test bench and subjected to vibrations which simulate working loads. The same method can be used to study deformation of parts on a running machine.

The strain gauge indications reveal the surfaces withstanding the highest tensile stresses, which consequently need strengthening. Small values or complete absence of stresses shows it is possible to lighten the part at these sections.

Internal stresses in the material are exposed in the following way: a strain gauge (sensor) of a measuring instrument is glued upon the surface area to be examined and the instrument set to zero; the section of metal to be tested is cut out and the internal stresses deduced from its size change.

Recently strain gauges with a base length of 0.5 mm have been manufactured. Also produced are semiconductor (silicon) strain gauges, whose sensitivity factor is 100-200 times higher than that of the constantan sensors and which have an elastic-plastic measuring range of up to 20%.

Fatigue tests are conducted with the aid of multichannel instruments which enable measurements to be taken simultaneously at many (up to 200) points. The cyclic stresses occurring at these points and varying between 50 to 50 000 Hz are recorded in digital or coded form on a film or tape. The obtained data can also be transmitted over a distance to an illuminated display, where the data is presented in the form of stress curves.

Temperature-compensated strain gauges employed for strain measurements under high temperatures exclude the effects of the apparent strains due to the thermal surface expansion. Such compensated constantan wire gauges allow temperatures up to 300°C to be measured, those of nichrome wire, up to 750°C and of platinum wire, up to 1100°C. High-temperature strain gauges are fixed on the tested surface by means of a ceramic cement.

The *lacquer-coating* method has the advantages of simplicity and visuality. The surface to be tested is coated with a thin film of lacquer. After a load has been applied to the specimen, in the zones of high strain in the lacquer coating there will form cracks perpendicular to the direction of tensile stresses. This enables stress directions to be defined. If loads are applied gradually and the ultimate strength of the lacquer is a strain function known, then from the appearance of the first cracks the deformation (and stresses) may be determined.

As the load increases, the cracks widen and, at the same time, new cracks appear in the regions where stresses begin to exceed the ultimate strength of the lacquer film. The final pattern of the cracks show the tensile stress distribution over the area being researched.

In compression zones the orientation and magnitude of compressive stresses can also be deduced from the wrinkles and creases occurring in the lacquer film. Further compressive stress increases turn these wrinkles and creases into cracks.

One of the simplest formulae of the lacquer composition reads: 60 g resin and 10 g celluloid dissolved in 100 g acetone. By varying the composition a spectrum of lacquers with different strength characteristics may be obtained, thus increasing the application and accuracy of the method.

Surface strains occurring under high temperatures can be determined through the agency of brittle ceramic coatings deposited by hotpulverization upon the surface to be tested.

The low sensitivity and the impossibility of quantitatively measuring stress values are disadvantages of the film technique: films crack only under sizable strains, which belong to the category of plastic deformations. To observe elastic deformation by the method is difficult. The film-coating method by no means can replace other, more exact methods of stress measurement, for example, strain gauges. However, this simple method does allow the general character of the stress distribution to be quickly seen and, with some skill, weak and nonrigid sections to be faultlessly located and, if necessary, strengthened.

Moreover, it is a valuable aid when applying and positioning strain gauges, since it allows the stressed areas to be preliminarily selected.

The film method is used for studying stresses on a working machine. Like strain gauges, it only defines the amount of stresses on the specimen surface, which, in most cases, are of decisive importance for the component's strength.

The most simplest method of checking parts for strength and rigidity is by bench tests under a static load and conditions approximating most closely the working ones. The resultant deformations are measured by indicators or strain gauges.

Bench tests provide best results when checking such parts as high-speed rotors, e.g., rotating discs of centrifugal or axial compressors undergoing mostly centrifugal loads. The speed of the part being tested is gradually increased to a value, which exceeds by 20-40% its working speed (correspondingly increasing stress values 40-100% in excess of design stresses). Such tests enable actual loads to be simulated (except thermal stresses which develop in heat engine rotors).

To determine safety marging tests are sometimes carrier to overspeed limits, i.e., to complete failure.

The most trustworthy, although the most expensive way of testing, is a complex check of the machine as a whole. Such a testing includes a long-time trial running of the machine under severe working conditions, either on a bench or in the field. At definite intervals of time the machine is partially or completely dismantled and the conditions of parts evaluated with the aim of forecasting forthcoming failures. This method jointly reveals the machine elements which are inferior not only in respect of their strength, but also in wear-resistance. Reliability of parts is only established indirectly, namely, on their good condition after prolonged operation.

(g) *Raising the Design Stresses*

Decreases in the weight of a part can only be achieved by increasing the design stresses and decreasing the safety margins.

First an essential stipulation. We speak here about an actual safety margin decrease consisting in the increase of the actual stresses and reduction of the part's cross section. A formal decrease in the safety margin, obtained only as a result of correcting the stress values, is another matter.

Let us explain it by an example.

The safety margin

$$n = \frac{\sigma_b}{\sigma} \tag{3.21}$$

where σ_b = breaking stress;
σ = design stress

For the simplest case of a cylindrical part in flexure the design stress will be

$$\sigma = \frac{M}{0.1D^3}$$

where M = bending moment acting upon the part;
D = diameter of the part

Substituting this expression into Eq. (3.21), we get

$$n = \frac{\sigma_b 0.1D^3}{M}$$

The actual lowering of the safety margin will only occur if the part diameter is decreased. Let the shaft diameter be decreased by 10%. Then the new safety margin will be

$$n' = n \cdot 0.9^3 = 0.74n$$

and the new value of the design stress

$$\sigma' = \sigma \left(\frac{1}{0.9}\right)^3 = 1.35\sigma$$

With the formal correction of the design stress value Eq. (3.21) will have the form

$$n = \frac{\sigma_b}{a\sigma}$$

where a is a factor accounting for higher stresses when correcting the design stresses (for example, when considering stress concentrations).

Let a be equal to 2. Then, the safety margin is halved, but this, however, does not indicate any actual drop in the strength or rise in the stresses. One and the same part, calculated in different ways, may have different safety margins, but nevertheless fail under the same breaking load, regardless of the design safety margin.

Reduction of weight through a decrease in the safety margin depends on the character of loading. The largest weight savings are obtained in the simple tension-compression case. The dependence of weight on stress is expressed by the formula

$$\frac{G}{G_0} = \frac{\sigma_0}{\sigma}$$

where G_0
and σ_0 = original weight and stress, respectively;
G = weight at the increased design stress σ

In flexure and torsion the relationship between weight and stress is less strong

$$\frac{G}{G_0} = \left(\frac{\sigma_0}{\sigma}\right)^{2/3}$$

It should be borne in mind that increasing design stresses without changing the part shape is always accompanied by lower rigidity, which in many cases defines the serviceability of the part.

In terms of effectiveness, the method of increasing the stress level concedes to other ways of decreasing weight.

Let us consider it by an example. Assume that a shaft is loaded with a transverse flexural force or torsional moment. If the design stress is significantly increased (for instance, 1.5 times) the shaft outside diameter can be decreased in the ratio of $1.5^{-\frac{1}{3}} = 0.87$. In doing so, the shaft weight is reduced in the ratio of $1.5^{-\frac{2}{3}} = 0.75$ (i.e., it is decreased by 25% only), and its rigidity, in the ratio of $1.5^{-\frac{4}{3}} = 0.57$ (by 43%).

Such a weight reduction can be obtained by drilling in the shaft an axial hole whose bore is 0.5 that of the outside diameter. In this case the stresses will rise by only 6% and the rigidity, fall also by 6%.

A weight saving of 25% at the same stress level with rigidity increased by 5% can be obtained by enlarging the shaft outside diameter by as little as 5% and simultaneously increasing the bore diameter by 10%.

Thus, we conclude that giving rational forms to components is a much more effective and advisable way of reducing weight than enhancement of stress levels.

In most engineering constructions increasing stress has an insignificant effect because of the limited number of the calculable parts, whose weight is generally only a small fraction of the total construction's weight. The overwhelming part of constructions are incalculable parts. For the majority of machine classes (piston engines, compressors, turbines, pumps, machine tools, etc.) the

weight of the housings (generally castings) generally makes up 60-80% of the total machine's weight, and the calculable parts share 10-20%. If we take into account the fact that housings have high safety factors (due to manufacturing technology), then it will be quite obvious that the main potentialities for weight reduction are in these parts.

Assume that in a 5-t machine the cast housing components make up 70% of the weight and that the average wall thickness is 12 mm. Even if we reduce the wall thickness by 1 mm only, it will give us a weight-saving of approximately 300 kg. This result is unattainable even through the most scrupulous design calculations aimed at saving weight.

Naturally, lowering the weight of housings must not lower their strength, rigidity and stability. Reduction of cross-sections must be compensated for by improved casting technology, increased wall strength, elimination of casting defects, etc. Purely design techniques, which enable housing components to be lightened without deteriorating their strength and rigidity, include:

smoothing outlines; rounding corners and removing sharp edges; using shell-shaped and arched structures; proper ribbing pentroding; braces, struts, trusses, stays and other ties, binding the elements of a structure; using rational power schemes.

There are, however, some structures, in which calculable parts account for a comparatively large portion of the total weight. These include: machines, in which metal frameworks are predominant (jibs, gantry cranes, derricks); aircraft hull and wing structures with extensive carrying framework elements; frame constructions for various applications (support structures, uprights, towers, poles, etc.). For machines and structures of these types, refined calculations and reasonable safety factors give substantial weight savings.

(*h*) *Design Stresses and Safety Margins*

There are two main ways of selecting the design stresses and safety margins.

The first method (now rather obsolete) consists in the preliminary choice of the safety factor and the establishment of the design stresses on its bases, determining cross-sections and inertia moments from strength of material and theory of elasticity formulae accounting for the main loads in the design scheme (usually, maximum power or maximum revolutions).

The method is also applied in the reverse sequence: first the part sizes are roughly specified, then the stresses acting at critical sections are calculated, and finally the safety factor found. If the latter corresponds with established traditional values, the design is considered complete; if no, the part dimensions are corrected.

In this method, all factors determining deviations of true stress values from the designed ones are summarily inserted into the safety factor which consequently acquires a large value (5-10).

The second, modern way is aimed at complete and exact explanation for the actual stresses existing in the part. To help the analytical stress determination experimental methods are enlisted. Combination of the analytical and experimental methods establishes a more accurate stress distribution and determines stress values closer to the true maximum ones. With the improvement and refinements in the calculation methods the number of unknown factors reduces and the number of defined factors increases.

Indeterminate factors include internal stresses caused by macro- and micro-structural defects, and also the stresses due to imperfect manufacture and assembly. It is necessary to consider these factors when defining safety margins.

Furthermore, the safety margin must reflect the importance of the component part and possible consequence of its failure. If the failure of the part is likely to cause complete machine failure, then the safety margins must be increased.

The method of ascertaining stress values and putting into the safety factor a few random and unaccounted for factors is most correct. Naturally with such a refined method the safety factor may be lowered on average 1.5-3 times.

However, accurate computing methods are drawn up only for a limited number of loading cases and types of parts.

The third intermediate way tries to make up for deficiencies of modern design techniques by transferring unknown values into the margin of safety, but only in a differentiated form.

The margin of safety is generally expressed as the product of a number of individual factors, each representing some indeterminancy. The safety factor is often determined as

$$n = n_1 n_2 n_3$$

where n_1 = factor accounting for the difference between the values of the actual and theoretical loads, as well as the difference between the actual and theoretical stresses (due to errors in formulae);

n_2 = factor accounting for the heterogeneity of material, as well as for the influence of macro- and micro-defects and residual stresses in the material;

n_3 = factor accounting for the importance of the part and the reliability requirements for the part during operation

Some of the authors differentiate still more, asserting that the safety margin should be expressed as the product of many individual factors (ten and even more), thus encompassing all or nearly all

the indeterminate factors listed above (see p. 184). Then they give recommendations on the choice of numerical values of each of these factors, depending on the authenticity of calculations, quality of manufacture, complexity of the parts shape, etc.

It is evident that this system is similar in principle to the old system of gross safety factor. The difference is that when applying the former system the designer commits one gross error in the choice of the safety margin but applying the differentiated system he makes a few small mistakes which superimpose one upon another.

Obviously, the evaluation of the indeterminate factors by this method is hypothetical, causing, for example, a doubt factor as to the degree of calculation accuracy. The numerical evaluation of this factor presumes the existence of an absolutely true calculation allowing true stress values to be defined. Then there is no need for a correction factor: it is sufficient to introduce the true stress values into the calculations.

Furthermore, it is a well established fact that numerical values of correction factors, belonging to sharply different categories, as, for example, accuracy of calculation and perfection of technological processes, are incompatible.

In practice, however, with such calculations the use of the differentiated factor system often leads to adjustment of their numerical values in order to obtain admissible values of the total safety factor in the former conception of the term.

Thus, one may state that at present this problem has yet to be solved. The old methods are obsolete, and the new methods are not everywhere fully developed. Whenever exact experimentally proved methods of stress calculations are available, it is better to use the second method, and introduce into the safety factor only actual indeterminate factors.

When it is essential to use simplified methods of calculation they must be supported by experience of similar constructions already in operation.

Long trouble-free running of a machine is the best indicator that the stresses in the machine parts are acceptable (although it does not imply that these stresses cannot be decreased). Preservation of geometrical similarity between the part being designed and its prototype and selection of its absolute sizes from conditions of stress similarity from the main acting loads and perhaps of slightly higher stresses almost faultlessly lead to the development of a durable component part.

As a general rule, great care must be taken when decreasing safety factors. In most cases, the saving of weight at the expense of increased stresses is not great because of the relatively low weight of calculable parts in the structure of the majority of machines. The risk is great as in the first place the rigidity of parts is lowered

which in many cases determines the machines durability. Reduced rigidity may cause the appearance of supplementory, difficult to consider loads, which worsen working conditions for the parts. That is why when increasing design stresses it is obligatory to analytically or experimentally check the degree of reduced rigidity. It is better to combine increased design stresses with constructive methods of improving rigidity by giving parts rational forms.

An indispensible condition for the immediate comparison of safety factors accepted in different mechanical engineering branches is that identical calculation methods and identical strength of material theories must be assumed as the calculation base for complex stressed states.

It is also necessary to consider the specific branch of engineering. High-class machines manufactured under conditions of a strict technological discipline with carefully formulated quality control eliminating the possibility of defective material being passed for assembly may have lower safety factors. To mechanically give such low safety factors to other machines produced in less strict conditions would be a mistake.

In aircraft constructions the margins of safety related to the stresses which have been carefully computed by special methods and tested experimentally come to 20-30%. Obviously, such safety factor values are unacceptable for parts designed by routine simplified methods, manufactured and inspected less rigorously, and intended for a much longer working life than that in aviation.

(*i*) *Design Operating Conditions*

A mandatory condition for a correct design is the refining of the design operating conditions by a careful study of possible overloads which may occur during operation. The design operating conditions do not always coincide with those of the maximum power or speed. They may, for instance, include starting conditions, when some machines (ac squirrel-cage motors or d-c series-field motors) develop increased torques, or the conditions of braking, stopping, reversing, speed or load changing, or, finally, those of an abrupt load drop when the machine begins to overspeed.

In a machine whose drive incorporates an irreversible mechanism (worm gearings) higher stresses occur during stopping, when the rotating and rectilinearly moving driven links, owing to the stored energy, become driving elements in respect to the irreversible mechanism.

In crankshafts of internal combustion engines maximum stresses arise during torsional vibrations, while in flexible turbine shafts they occur when passing through the critical rotational range.

In many cases overloads can be eliminated or materially lessened through appropriate design modifications, e.g., by introducing regulators or speed limiters, limit speed monitors or clutches, vibration dampers, etc.

Under certain conditions overloads during machine operation are inevitable. This is the case, for example, in road-building machines, used on heavy or rocky terrains, damp soils, slopes, side banks, etc. For processing machines, variations in the quality of raw materials being handled is of great importance.

All these factors should be carefully studied from the standpoint of their influence upon strength before finally selecting the design scheme.

3.5. Materials of Improved Strength

An effective means of lowering a construction's weight is to use stronger materials. In contrast to the increased stress method, accomplished by lowering the safety factor with the risk of weakening the part, reliability in this instance is not impaired (if the same value for the safety factor is preserved). Another dif erence is that the improved material strength method can be applied to all parts without exception while the increased stress method is valid only for calculable parts.

The principal methods which enable materials to be strengthened include: hot working, alloying, thermal and thermal-chemical treatment, cold working.

Hot working under pressure. Strengthening of metals through hot working occurs due to the conversion of a loose ingot structure into a compact one with oriented crystallites. As the liquid metal cools, voids appear at the boundaries of the forming crystals. The voids are caused by metal contraction in the transfer from the liquid to the solid state, by the evolution of gas bubbles as the solubility of gases in metal decreases with decreasing temperature, etc.

As crystals are growing, the impurities, which are inevitably present in metal, are ejected and accumulated at the grain boundaries, where crystallization is completed. Because of this cast structure, insufficient bond between grains is inherent causing the lowered strength and toughness of cast metals. In the course of hot working (by pressure) the voids amidst crystallites are reduced and welded, while interlayers of impurities between crystals are crushed and, under the effect of high temperature and pressure, pass into metal.

For improving metal strength the recrystallization process is of great value. Recrystallization takes place during metal cooling within definite temperature interval (450-700°C for steels). From the crystals fragments, which are crushed and deformed during reduction, are formed new finely-sized grains and the impurities in the growing recrystallized grains remain absorbed. For this reason the structure

of forged metals consists typically of fine rounded grains closely associated with one another, which accounts for the increased strength and toughness of forged metals.

Forged and, particularly, rolled metals show anisotropy of mechanical properties along and across the grain. Toughness is especially influenced by the direction of metal fibres (Fig. 90).

The direction of metal fibres in forged and stamped parts should agree with the configuration of the part and the direction of acting loads. Crankshafts, forged or stamped in several passes, with fibres following the throw configuration (Fig. 91*b*), possess much higher strength than crankshafts, made of prismatic blanks, with sheared fibres (Fig. 91*a*). The gear produced by hot rolling plus

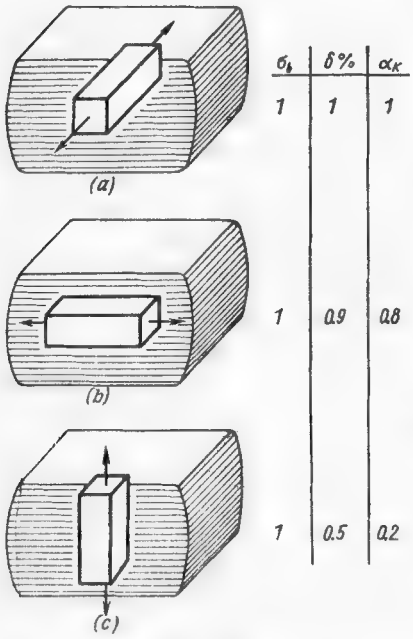


Fig. 90. Ultimate tensile strength σ_b, elongation δ and specific toughness α_k of steel as influenced by orientation of fibres

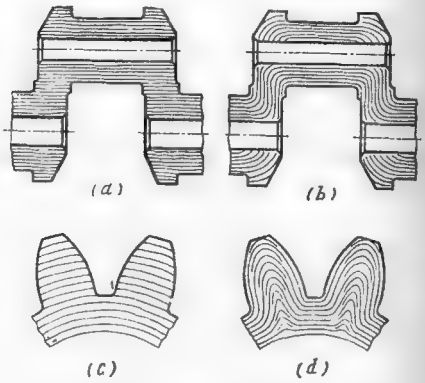


Fig. 91. Orientation of fibres

cold sizing will have correct grain orientation with respect to the loads acting on the gear teeth (Fig. 91*d*).

Alloying has various aims: to improve corrosion resistance and heat resistance; to give better welding properties and special physical characteristics. The main purpose of the alloying process is to improve strength with a differentiated improvement in other particular characteristics of the strength, toughness, plasticity, elasticity and wear resistance. Addition of certain elements may improve hardenability of steels, thus allowing better mechanical properties throughout the entire cross section of a part to be obtained.

The best all-in-all strength characteristics are possessed by chrome-nickel steels, particularly polyalloy chrome-nickel-tungsten and chrome-nickel-vanadium steels.

To acquire good mechanical properties it is necessary to complement alloying with thermal treatment.

The comparative characteristics of alloy and carbon steels are given in Table 7.

Table 7

Average Strength Characteristics of Carbon and Alloy Steels
(with optimum heat-treatment)

Steels	Tensile strength, σ_b, kgf/mm²	Yield limit, $\sigma_{0\,2}$, kgf/mm²	Elongation per unit length, δ, %	Fatigue limit, σ_{-1}, kgf/mm²	Impact strength a_k, kgf·m/cm²
Low carbon	35-50	25	25	20	3-6
Medium carbon	60-80	40-50	12	25-30	4-8
High-strength, alloy	100-180	100-150	6-8	60-100	6-10

Strengthening thermal treatment (hardening with high, medium and low tempering, isothermal hardening) causes the formation of unbalanced structures with strongly deformed crystal lattices (sorbite, troostite, martensite, bainite).

The composition of these structures, the sizes and shapes of grains, as well as different mechanical properties are obtainable by regulating the conditions of heat treatment. For constructional steels the application of quenching and tempering heat treatment (hardening with high tempering) assures the most favourable combination of strength, toughness and plasticity.

In recent years the induction hardening process has become very popular, which consists in heating a component's surface layers with high-frequency currents. Besides its purely technological advantages (economy, high productivity) this kind of heat treatment has a significant strengthening effect resulting from residual compression stresses in the hardened surface layer.

Chemical-thermal treatment consists in saturating the surface layer with carbon (carbonizing), or nitrogen (nitriding, cyaniding), with the formation in the latter case of Fe-nitrides and alloy elements. These kinds of heat-treatment are aimed at making the surface hard and wear-resistant. At the same time they increase strength (particularly under the conditions of cyclic loads) owing to the formation of a compression stressed state in the surface layer.

Cold working (shot blasting, rolling, coining, cold reduction and sizing) promotes the development of compressive stresses in the surface layer and enhances the fatigue strength.

(*a*) *High-Strength Cast Irons*

Grey cast irons are one of the most widely used constructional materials. Cheapness, good castability, high resistance to cyclic loads—these qualities ensure wide application of grey cast irons in the manufacture of housing components (castings) of stationary and transport machines when weight considerations are of secondary importance.

Disadvantages of grey cast irons are their low strength, low impact toughness and brittleness.

Modification is the first measure in improving the strength of cast irons. This means adding small amounts of inoculants (calcium-silicon, ferrosilicon, graphite powder) to the liquid iron before casting. These inoculants (additives) improve cast iron properties and promote homogeneity of the cast structure in all sections. Graphitizing modification stops the formation of graphite flakes so that the latter are present as nodules preventing chilling and giving a pearlite structure, this being more favourable for the strength of this material. The inoculated irons have a strength level 30-50% higher than that of grey irons.

In recent years methods have been developed which produce high-strength irons, i.e., alloy cast irons (containing Mg, Mn, Cr and some other elements), heat-treated to give granular pearlite with globular-shaped graphite inclusions. The process is called spheroidizing modification (inoculants: Mg, Cr or alloys of these metals with Cu and Ni). A typical chemical composition of a high-strength alloy cast iron is: 3.4-3.6% C; 2-2.2% Si; 0.03-0.06% Mg; 0.15-0.25% Cr; 1.15-1.3% Mn; not more than 0.005% S and 0.12% P.

The heat treatment includes: normalizing at 950°C for 6-8 h with subsequent cooling at a rate of 30-60°C/min, then tempering by heating to 700-720°C for 8 h with subsequent cooling in air.

High-strength cast irons have better mechanical properties than grey cast irons (see Table 8), approaching, in this respect, steels. These high-strength cast irons are used for manufacturing heavily loaded housings with complicated configurations. They can be processed by induction hardening, shot-blasting and nitriding. High-strength nitrided cast irons (with admixture of Al) possess a hardness of about 900 VPH.

Nowadays, important heavily loaded parts are made from high-strength cast irons, e.g., crankshafts, which successfully compete in strength with forged and moulded shafts of carbon and low-alloy

Table 8

Characteristics of High-Strength and Grey Cast Irons

Characteristic	Irons	
	grey	high-strength
Ultimate strength, kgf/mm²:		
tensile σ_b	15-30	45-80
flexural σ_f	30-45	50-90
Yield strength $\sigma_{0.2}$, kgf/mm²	10-20	40-50
Fatigue strength σ_{-1}, kgf/mm²	6-15	15-25
Elongation per unit length δ, %	<0.3	2-10
Impact strength α_k, kgf·m/cm²	0.2-0.4	1.5-3
Elastic modulus E, kgf/mm²	8000	15000

steels and even exceed the latter in respect of wear-resistance. Furthermore, the manufacturing costs of cast iron shafts are many times less than those of forged steel shafts.

Castability (casting properties) of high-strength cast irons is worse than that of grey cast irons (shrinkage of grey cast irons is 0.8-1.2% and of high-strength cast irons, 1.3-1.8%). However high-strength cast irons cast better than casting steels. Particular attention should be paid to desulfurization of cast irons, otherwise magnesium sulfides will occur in castings (in the form of the so-called black spots), which will cause local softening of castings.

It is noteworthy that in respect of cyclic toughness the high-strength cast irons are much more inferior to the grey cast irons.

By cyclic toughness is meant the property of metals to partially transform the energy of elastic strain into heat as a result of internal losses due to friction. The higher the cyclic toughness, the better the metal property to dampen vibrations when subjected to cyclic loads. Grey cast irons possess the highest cyclic toughness.

The value of cyclic toughness is given by the hysteresis coefficient Ψ (percentage ratio of energy loss v per strain cycle to the full strain energy w):

$$\Psi = \frac{v}{w} \cdot 100\%$$

Figure 92 gives hysteresis coefficient values for cast irons and steels as a function of amplitude τ of stress variations per strain cycle. As evident from the diagram, the cyclic toughness of grey cast irons is 5-6 times that of carbon steels and 10-20 times that of

alloy steels. High-strength cast irons approximate steels in terms of cyclic toughness and inoculated cast irons are midway between the grey and high-strength cast irons.

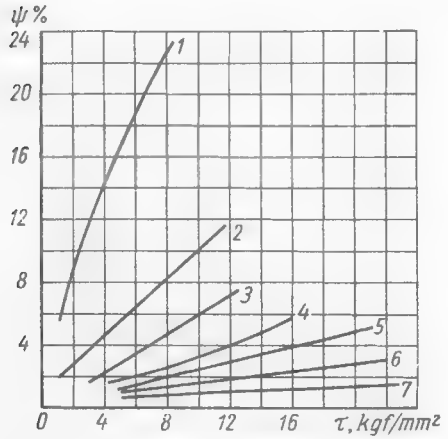


Fig. 92. Hysteresis coefficient Ψ as a function of stress amplitude τ per deformation cycle

1 — grey cast iron; *2* — inoculated cast iron; *3* — steel grade 20; *4* — steel grade 45; *5* — high-strength cast iron; *6* — steel casting; *7* — steel grade 40X

The cyclic toughness of non-ferrous metals is extremely low, except magnesium alloys which approximate carbon steels in their cyclic toughness.

(*b*) *Extra-High Tensile Steels*

The development of superstrong materials is based on the modern concept of dislocations (i.e., local distortions of atomic-crystalline space lattices,), which are considered to be the original cause of discrepancy between the actual strength of metals and the theoretical value, predicted on the basis of atomic bonds in crystalline lattices. The theoretically inferred strength value amounts approximately to (0.1-0.15) E, where E is the Young's modulus. Yet, the actual strength is tens, and occasionally hundreds times less than theoretical values. In other words, in modern metals only a very small portion of their potential strength is utilized.

Up till recently it has been a generally accepted consideration that the process of plastic strain consists in simultaneous shear of crystal planes one relative to another. However, such a concept is inconsistent with the magnitude of those forces which are needed to overcome the strength of atomic bonds at the slip planes. Now it is generally considered that the shear occurs not at once, but in subsequent stages.

In the areas of dislocations, as a result of crystal lattices distortion, there are formed plateaus of easy slip, so that a comparatively small shearing force will suffice to displace crystal planes in the area over an interatomic distance.

Such a displacement will result in a corresponding shift of the plateau along or opposite to the direction of the acting force. In the new location of the plateau another displacement over an interatomic distance will occur, this being, in turn, followed by a new shift of the plateau.

In this way the plateau will successively move in the direction of the acting force, making, in doing so, the entire crystal plane shift over an interatomic distance. Should the force be continued, the action will keep on repeating many times so that a macroshift of crystal planes will take place.

Obviously, successive shearing, which requires only a local rupture of atomic bonds, will be actuated by such a force that is many times less than that necessary to simultaneously displace the entire crystal plane.

It is the mechanics of the described shift which is mostly responsible for the lowered actual metal strength when compared with theoretical values.

The displacement of ready-to-slip plateau will continue until the dislocation has emerged onto the surface of a crystal block or encountered some obstacle (e.g., foreign interstitial solute, perpendicular dislocation, similar dislocation but of the opposite sign, etc.). Dislocations of opposite signs mutually cancel each other.

Thus, we may conclude that increasing heterogeneities, i.e., enlarging the amount of admixtures and the number of crystal lattice distortions, as well as comminuting crystal blocks, will make metals stronger owing to impeded and locked dislocations.

Dislocations inevitably occur in huge quantities in all metals. For instance, the average density of their distribution in steels reaches 10^8-10^{10} cm^{-2}.

Dislocations arise due to many reasons: superfluous interstitial crystal layers, the so-called extraplanes (line dislocation), spiral displacement of crystal planes relative to each other (screw dislocation). An alternative of dislocations is a vacancy, i.e., absence of atoms in the nodes of crystal lattices, as well as foreign atoms inoculated into interstices. Local distortions in lattices occur under application of external loads and also in the zones where internal stresses act.

The existing dislocation may cause new dislocations in adjacent areas. There are sources of spontaneous dislocations: two superimposed line dislocations will form a continuously acting generator of dislocations (Frank-Read sources).

There are two main methods of increasing strength of metals:

(1) elimination or reduction of the number of dislocations (production of metals possessing homogeneous and regular crystalline structure);

(2) increasing the number of heterogeneities (creation of obstacles that will check the development and propagation of dislocations).

The potentialities of the first method are rather limited, since a defectless structure can only be obtained for extremely pure materials and in very small volumes, which would exclude the inception

and development of dislocations. Exceedingly thin whisker crystals have been prepared under laboratory conditions. Such a whisker crystal (several millimetres long and 0.05-2 μm thick) has a tensile strength of 1350 kgf/mm², this being approximately 100 times that of conventional commercial iron and 10 times that of high-grade alloy steels. Alongside such a high tensile strength the whiskers possess good elasticity characteristics. Thus, the elastic elongation of whiskers reaches 5%, while the same parameter offered by conventional iron does not exceed 0.01%.

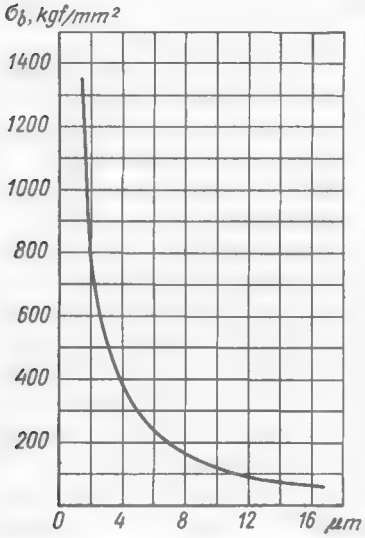


Fig. 93. Strength of iron whiskers (kgf/mm²) as a function of diameter (μm)

Enhanced strength and elasticity of single crystal whiskers are obtained owing to extreme purity of their materials and regularity of crystal structure. The development of dislocations in whiskers is practically impossible, since the diameter of a whisker is less than the average extension of a dislocation. With an increase in diameter the strength of crystal whiskers sharply falls (Fig.93) because of dislocations.

The whiskers are also produced from non-metallics (graphite aluminium oxide Al_2O_3, silicon oxide SiO_2, silicon carbide SiC); the strength of non-metallic whiskers is higher than that of their metallic counterparts (Fig. 94).

The actual strength of whiskers is 50-60% of the theoretical values. The tiny dimensions hamper, however, their technical application.

Perhaps, the only real way of whiskers utilization is their employment in the manufacture of composite materials, comprising single crystal whiskers, laid in a definitely oriented pattern within a metallic (e.g., aluminium) or plastic matrix. If the whisker has sufficient length to ensure a good bond with the matrix (along its side surfaces) its strength will be successfully used. It is noteworthy that the strength of composite materials, which contain 40-50% (by weight) of whiskers, attain approximately 30% of the latter's strength. Thus, a composition of sapphire single crystal whiskers (Al_2O_3) and metallic aluminium has a strength 500-600 kgf/mm².

Naturally, such materials are very expensive (their cost nearly equals that of platinum); for this reason, their application is limited to only special constructions.

The second method seems more promising, tending to increase the degree and number of heterogeneities. The first step is alloying and heat treatment, whose strengthening effect, in essence, amounts to an increase in the density of dislocations.

Further achievements in the development of high-tensile steels are based on the fact that in some multi-component alloy steels (comprising a relatively low percentage of alloying elements) the process

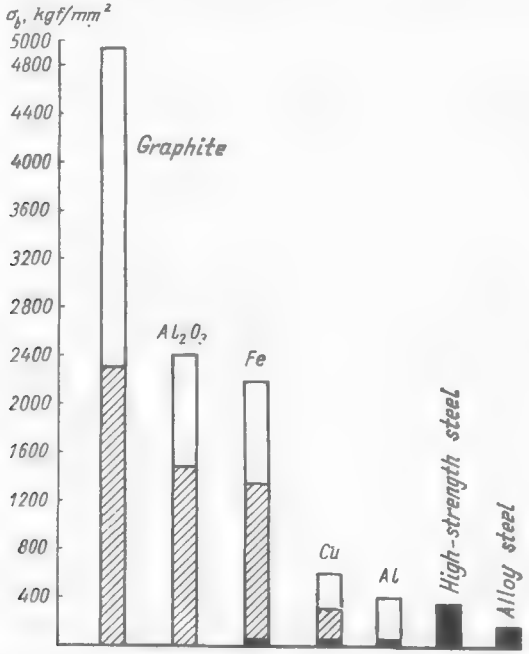


Fig. 94. Theoretical strength of materials (unshaded rectangles), strength of whiskers (shaded rectangles) and actual technical strength (black rectangles)

of cooling, starting from the temperature of austenitic transformation within a certain range of temperatures (450-550°C), is not accompanied by decomposition of austenite (it should be pointed out that such a decomposition is inevitably followed by the formation of hard ferrite-cementite mixtures). Consequently, within this range of temperatures steels can remain plastic for an unlimited period of time and be ready for forging, stamping and rolling.

Thus begins the thermomechanical processing method combining, in fact, the processes of thermal treatment and plastic deformation.

Low-temperature thermomechanical processing (LTTP) is an intensive plastic deformation of steel within the temperature range of the stable austenitic state.

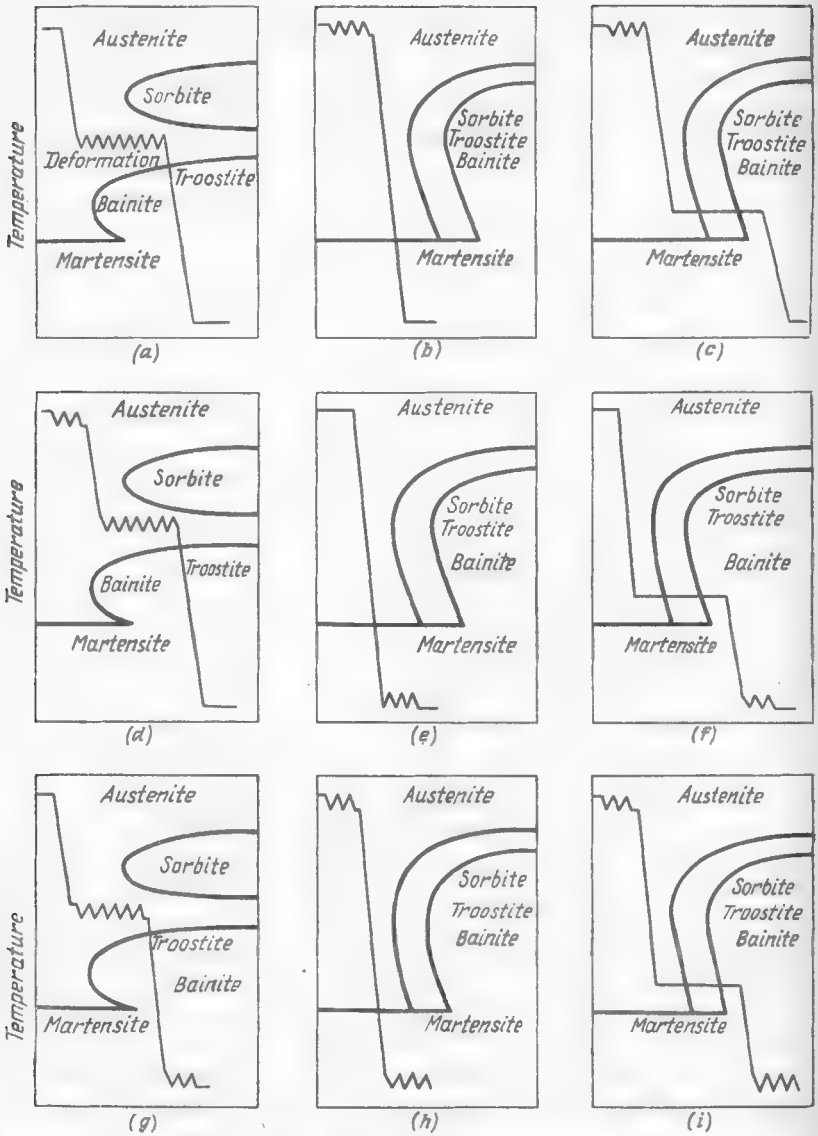


Fig. 95. Schematic representation of thermomechanical treatment (diagrams of isothermal austenite decomposition: temperature — time)

The process (Fig. 95*a*): heating to 900-1000°C; rapid cooling down to 450°-550°C; multiple plastic deformation at this temperature with a high degree of deformation (90%); martempering (to produce a fully martensitic structure); and final tempering at 250-400°C.

The low-temperature thermomechanical treatment can readily be applied to steels having the following approximate chemical composition: 0.4-0.6% C; 1-1.5% Ni; 0.7-1.5% Mn; 1-1.5% Si; 1-3% Cr and 0.5-1.5% Mo. The steels of such a composition have the stable austenitic state within the above-mentioned range of temperatures.

The LTTP assures a significant increase in the material strength (tensile strength $\sigma_b = 320\text{-}350$ kgf/mm^2, yield limit $\sigma_{0.2} = 280\text{-}300$ kgf/mm^2 with elongation $\delta = 8\text{-}12\%$). This is about twice the strength indices of the best modern alloy steels. Of great significance is the fact that the LTTP sharply increases fatigue strength.

The enhancement of strength attained by LTTP is attributed, in the main, to the substantial destruction of the crystalline structure as a result of semi-plastic deformation, which is accompanied by comminution of crystalline groups (blocks) down to a fourth-fifth fraction of crystalline groups obtainable by conventional heat treatment.

The shortcoming of the LTTP lies in that the treated parts cannot be subjected to high temperatures because the steel will lose in this event that hardness which it has acquired before. Hence, the LTTP-treated parts forbid welding.

The process is applicable for rolled stock and parts of simple form. Intricately-shaped components will not yield good results when subjected to the LTTP, as it is impossible to obtain uniform deformation and homogeneity of metal properties throughout the entire component.

Another disadvantage is the necessity for increased forces which are needed to deform the material in its semiplastic state.

To avoid this disadvantage use is made of the *high-temperature thermomechanical processing* (HTTP), by means of which (Fig. 95*b*) the material is deformed in the range of temperatures between 800-900°C, with the deformation degree reaching 20-30%. Afterwards the component is martempered and after-hardened. Occasionally heat treatment for bainite takes place (Fig. 95*c*).

The HTTP method brings about a lesser accretion in strength as compared with the above described method (LTTP): the strength level is raised to 220-280 kgf/mm^2, yet it is still 1.5-2 times that obtained by applying separately pressure working and heat treatment. Furthermore, the HTTP technique betters plasticity and impact toughness in parallel with reducing the sensitivity of steel to stress concentrations.

The HTTP treatment may be applied, though with poorer strength results, to conventional medium carbon steels. Thus, high temperature thermomechanical processing increases the ultimate strength of steel grade 45 to 180-200 kgf/mm².

When combined, the LTTP and HTTP treatments (Fig. 95*d*) will produce a tensile strength increment of 15-20%.

Still another method of material hardening is based on martensite strain ageing (MSA). In this case (Fig. 95*e*) the steel is first subjected to the usual heat treatment (hardening plus tempering at 250-400°C) and afterwards deformed in a cold state, the deformation degree being 1-3%. Then follows ageing for 1-2 h at a temperature which exceeds by approximately 100°C the tempering temperature. In the process of ageing the steel tensile strength rises to 200-250 kgf/mm². It is significant that the ratio between the yield limit and ultimate tensile strength $\frac{\sigma_{0.2}}{\sigma_b}$ becomes approximately equal to 1. Owing to this the strain-aged steels, as regards their yield limit value which is the main strength characteristic of the material, are close to steels which have been strengthened by more complicated methods (such as those described above).

Deformation can be performed by any method: reduction, extension, torsion, stamping, rolling, etc. Intricately-shaped components are deformed by applying loads, which simulate the working ones. Thus, storage vessels are hardened by applying high internal pressure with subsequent ageing.

Increased hardness in the course of strain ageing is obtained on account of two simultaneously acting factors: cold working (i.e., higher density of dislocations) plus comminution of martensite blocks.

An alternative to this method is bainite hardening followed by strain ageing (Fig. 95*f*). Satisfactory results are obtained also by combining strain ageing with LTTP (Fig. 95*g*) and HTTP (Fig. 95*h*, *i*).

In recent times a hardening process has been suggested which is based on the ageing of carbonless alloy martensite. This method can be applied to carbonless (<0.01% C) alloys Fe — Ni — Co — Mo, containing 18-20% Ni; 7-10% Co and 3-5% Mo with obligatory admixtures of Ti (0.3-1.5%) and Al (0.1-0.3%), which are, in fact, the primary hardening elements.

These alloys are heat-treated by martempering, which, in contrast to conventional tempering of alloy steels, does not require high cooling rates and proceeds as the steel is being cooled down in open still air from 800-1000°C (generally alloys are hardened from forging temperature). This hardening will result in soft martensite (10-15 Rc), which readily yields to deformation in the cold state.

The material is then subjected to ageing, keeping it at 400-500°C for approximately 3 h. After ageing the ultimate tensile strength

increases to 210-250 kgf/mm² (with $\frac{\sigma_{0.2}}{\sigma_b} \approx 1$); the martensite hardness is raised to 50 Rc, keeping its high plasticity (δ = 10-12%) and toughness (a_k = 8-12 kgf/cm²). The strengthening effect is attained mainly due to segregation of intermetallics, such as Ni (Ti, Al) and Ni_3 (Ti, Al, Mo).

Martensitic-ageing alloys have good technological characteristics. Ageing does not distort the part, this treatment can be used at the final manufacture stage. Such alloys can be worked through hot plastic deformation of any kind (forging, rolling). In a hardened state (prior to ageing) these alloys can be worked by pressure (deep drawing, spinning operations, etc.). The alloys have good machining and satisfactory welding qualities after hardening and after ageing. Softening in the weld zone (when welding in aged state) is eliminated by reageing.

The disadvantage of the martensitic-ageing alloys is the increased content of expensive Ni and Mo. Ample tensile strength and good toughness can be obtained by introducing 1.5-2% Mn, the Ni content not exceeding 8-12%.

Soviet scientists have developed a method which enables low-carbon steels to be strengthened by multiple thermomechanical processing (MTMP). It is in deforming a specimen 5-6 times, each deformation stage corresponding to the yield plateau length on the stress-relative elongation diagram (total deformation reaching 6-8 per cent) until the yield plateau fully vanishes. This is followed by ageing at 100-200°C for 10-20 h. The processing allows the yield limit to be raised by 25-30% (nearly approaching the ultimate strength) and the fatigue limit by 30-50%.

Recently, along with the thermomechanical processing, additional strength is obtained by applying a magnetic field which on the strength of the well-known magnetostriction phenomenon changes crystal sizes. The stresses caused by magnetostriction will be added to the stresses produced during the previous thermomechanical process, this strengthening the steel still further (approximately by 10-15% when compared with the original strength). This is called the *thermo-mechano-magnetic processing* method (TMMP).

Samples of super-high strength steels have been produced under laboratory conditions. These samples have ultimate tensile strength as high as 400-500 kgf/mm², i.e., ten times that of carbon steels and 3-4 times that of modern alloy steels.

With the appearance of high-strength steels arises a series of new design problems since parts made from the stronger metals are less rigid. This is because the modulus of elasticity of each metal has a stable value and only slightly depends on heat treatment and contents (in normal quantities) of alloying elements. Since elastic strains are proportional to the ratio between the stress and elasti-

city modulus, then with increased stress values (which is the reason for the application of high-tensile materials) the strains will also increase proportionally and the rigidity will fall inversely proportionally.

This will be true only if the length of parts is assumed unchanged (the case of most applications). The linear dimensions of a construction are generally chosen to suit machine operational conditions. Thus, in power generators and converters these dimensions depend on the working capacities and specifications, e.g., in internal combustion engines these dimensions depend upon cylinder size, which is, in turn, dependent upon the gas operating pressure; in machine tools, on the workpiece size to be machined; in metallic frameworks, upon the length and the height of the construction. In all these cases the use of high-strength materials will affect only the cross section and not the length of a part.

A series of machines exist whose linear dimensions are dependent only on the strength of materials. To such machines are referred reduction gear units. In this instance the application of high-strength materials allows not only cross sections to be decreased, but also a proportional reduction in the length of separate parts and the overall unit dimensions as a whole to be obtained. In these cases the introduction of high-tensile materials does not lower construction rigidity.

Let us consider a case when the linear dimensions of a part are unchanged. Assume, there are two equallystrong (in tension) bars of the same length—one from carbon steel grade 45, with on ultimate tensile strength 50 of kgf/mm^2, and the other from a high-strength steel with an ultimate strength 500 kgf/mm^2. The rigidity of the latter bar is obviously 10 times less than that of the former.

Now let us evaluate the absolute deformation values.

Take as an example a connecting rod, length $L = 400$ mm, of an internal combustion engine. If the compressive stress developed in a conventional steel rod is 20 kgf/mm^2, the elastic deformation will be

$$\lambda = \frac{\sigma}{E} L = \frac{20}{21\,000} \cdot 400 \approx 0.4 \text{ mm}$$

Now, if we compare it with the compressive deformation in a connecting rod from the high-strength steel with a proportionally less cross-sectional area (for the equistrong conditions), we shall find that the deformation is much greater

$$\lambda = 4 \text{ mm}$$

In the tension-compression case no means exist of fighting rigidity reduction because with the given values of σ and E the deformation depends only on the cross-sectional area and is quite independent

of its shape. In flexure, torsion and longitudinal bending the rigidity reduction is greater, but in the given case there are some means of fighting this phenomenon.

Assume two equistrong bars of equal length and similar cross-sectional profiles made from the same steels as in the above example and subjected to flexure or torsion. Then, for the above case the rigidity of the bar made from high-strength steel will be less by $10^{4/3} = 21.5$ times.

Let us consider a numerical example. Assume a shaft 60 mm in diameter and 400 mm long, supported at its ends and loaded in the centre with a force P.

The maximum deflection of the shaft under the action of flexural moment ($M_{flex} = PL/4$)

$$f = \frac{PL^3}{48EI} = \frac{M_{flex}L^2}{12EI}$$

As $I = W\frac{D}{2}$, then

$$f = \frac{M_{flex}L^2}{6EDW} = \frac{\sigma}{E} \cdot \frac{L^2}{6D}$$

If the bending stress in the carbon steel is 20 kgf/mm², then

$$f = \frac{20}{21\,000} \cdot \frac{400^2}{6 \cdot 60} \approx 0.45 \text{ mm}$$

The deflection of the high-strength steel shaft with its proportionally less section will have a very large value

$$f' = 21.5 \cdot 0.45 \approx 10 \text{ mm}$$

Thus, the application of high-strength materials with full use of their strength reserves and reduction of cross section without a corresponding decrease in length may lead to a catastrophic drop in rigidity.

Generally the rigidity characteristic is improved by increasing diametral sizes of the part and making simultaneously its walls thinner. However, in the case under consideration, this would be useless: with the increased moments of inertia the resisting moments of the part are simultaneously increased and this is accompanied by decreased stresses. Consequently, this method lowers the level of stresses which conceals the main advantage of high-strength materials, namely, the possibility of raising design stress to obtain a corresponding weight saving. The advantage is only partly realized with a very large reduction of wall thickness (in general mechanical engineering the wall thickness would only be 1-2 mm), i.e., this means a switch over to shell-like components.

For some engineering components (disks, containers, gears, connecting rods, levers, shafts, etc.) the shell-like shapes are practicable, although they require radical changes in design and manufacturing processes. Therefore, along with the increase in inertia moments it is necessary to apply other methods of lessening deformations, such as shortening the length of parts, closer arrangement of supports, etc.

In all cases the employment of high-strength materials puts new problems before designers and technologists, the solving of which requires a lot of creative and original thought.

A positive feature of parts made from high-strength steels is their extraordinary high capability of withstanding impact loads being accounted for by the large elastic deformation values. The resistance to impact loads is approximately proportional to the ratio $\frac{\sigma_{0.2}^2}{E}$ [see Eq. (3.33)], where $\sigma_{0.2}$ is the yield limit and E, Young's modulus. Should the yield limit be considered as proportional to ultimate strength, then the resistance of high-strength steels to impact loads will be greater than that of conventional steels in the ratio

$$\left(\frac{\sigma_b'}{\sigma_b}\right)^2$$

where σ_b' and σ_b are the ultimate tensile strengths of the high-strength and conventional steels, respectively.

When $\frac{\sigma_b'}{\sigma_b} = 10$ the resistance of high-strength steels to impact loads will be 100 times that of conventional steels.

3.6. Light Alloys

The use of materials with low specific weights is a significant way of lowering the weight of constructions. However, real weight savings not only depend on the value of the specific weight; they also depend on the material strength. The lower strength and rigidity of the light materials may in a series of cases reduce to tought the weight savings.

Additional factors limiting the application of light-weight materials are: low hardness, insufficient resistance to corrosion, low heat- and cold-resistance, high sensitivity to stress concentrations. Finally, it is necessary to consider cost and availability of these materials or components made of them.

Some of the light-weight materials being applied in general engineering are:

	γ in kgf/dm³
aluminium alloys	2.6-3.2
magnesium alloys	1.8
titanium alloys	4.5
plastics	1-1.8
improved wood	1.3-1.6
ceramic materials	2.2-3.2

(*a*) *Aluminium Alloys*

Aluminium alloys are the most widely used light alloys. They are noted for their low specific weight ($\gamma \approx 3$ kgf/dm³), high heat-conductivity ($\lambda = 100\text{-}150$ cal/m·h·°C), satisfactory strength and plasticity and are easily machined. Many of them can be welded by the argon-arc method when using non-consumable tungsten electrodes or by hydrogen-arc welding. Use can be made of gas submerged welding (LiCl, NaCl, KCl, KF). Sheet materials are resistance-welded.

Aluminium alloys withstand corrosion in dry atmospheres, are resistant to the influence of alkalies and weak acid solutions, but corrode in humid (especially sea) air and are non-resistant to strong acids and rather soft (their hardness varies within 60-130 BH). Coefficient of linear expansion $\alpha = (20 \text{ to } 26) \cdot 10^{-6} °C^{-1}$. Young's modulus $E = 7000\text{-}7500$ kgf/mm².

The strength of aluminium alloys rapidly decreases with the increase of temperature. There are, however, some alloys which retain satisfactory mechanical properties up to 250-300°C.

Aluminium alloys are divided into two main categories: *castable* and *wrought* (i.e., worked by forging, stamping, rolling).

Castable alloys (Table 9) by their chemical composition are subdivided into aluminium-copper, aluminium-magnesium, aluminium-zinc, and also aluminium-zinc-silicon, aluminium-copper-silicon, aluminium-silicon and composite (with admixtures of Ni, Ce, etc.).

The highest combined characteristics belong to the aluminium-silicon alloys (*silumins*), distinguished for their low specific weight ($\gamma = 2.6\text{-}2.7$ kgf/dm³), good casting and welding qualities and high corrosion resistance. Silumins can be used to particularly good advantage for casting thin-walled intricately shaped components. To enhance their mechanical properties silumins are generally inoculated before casting (with metallic sodium, Na and K fluorides), owing to which the silicon inclusions acquire a grain-like structure favourable to strength.

Aluminium alloys are often used for casting bases and housings. The low strength and rigidity of aluminium alloys are compensated for by increasing cross-sections, moments of inertia and resisting.

Table 9

Chemical Composition and Mechanical Properties of the Main Castable Aluminium Alloys

Alloys	Alloy grade	Chemical composition, %					Mechanical properties	
		Cu	Mg	Mn	Si	Zn	Ultimate tensile strength, σ_b, kgf/mm²	Elongation, δ, %
Al—Cu	AJI7	4-5	—	—	—	—	20	6
	AJI19	4.5-5.3	6-1.0	—	—	—	30	8
Al—Mg	AJI8	—	—	9.5-11.5	—	—	29	9
Al—Zn—Si	AJI11	—	—	0.1-0.3	6-8	7-12	18-25	1.5-2
Al—Cu—Si	AJI5	1-1.5	—	0.35-0.6	4.5-5.5	—	16-23	0.5-1
	AJI6	2-3	—	—	4.5-6	—	15	1
Al—Si	AJI2	—	—	—	10-13	—	14-16	1-4
	AJI4	—	0.2-0.5	0.17-0.3	8-10.5	—	15-24	1.5-3

Table 10

Chemical Composition and Mechanical Properties of Some Wrought Aluminium Alloys

Alloys	Chemical composition, %							Mechanical properties			
	Cu	Mg	Mn	Ni	Fe	Si	Other elements	Ultimate tensile strength, σ_b, kgf/mm²	Yield limit, $\sigma_{0.2}$, kgf/mm²	Endurance limit, σ_{-1b}, kgf/mm²	Elongation, δ, %
Д1	3.8-4.9	0.4-0.8	0.4-0.8	—	—	—	—	40-45	25-30	10-12	10-12
Д16	3.8-4.9	0.3-0.1	1.2-1.8	—	—	—	—	45-50	25-35	12-15	8-10
В95	1.4-2.0	0.2-0.6	1.8-2.8	—	—	—	5-7Zn 0.1-0.25Cr	50-60	40-50	12-15	5-7
АК2	3.5-4.5	0.4-0.8	—	0.8-1.3	0.8-1.3	0.5-1.2	—	40-45	25-30	10-12	4-5
АК4	1.9-2.5	—	1.4-1.8	1-1.5	1.2-1.5	0.5-1	—	35-40	20-25	8-10	6-8
АК6	1.8-2.6	0.4-0.8	0.4-0.8	—	—	0.7-1.2	—	35-40	20-25	8-10	5-6

Despite this the employment of aluminium alloys gives significant weight savings.

Components made from aluminium alloys which require sealing (e.g., sumps, oil-casings, boxes, etc.) are impregnated with synthetic thermosetting compounds (often with bakelite) and are then heated to the bakelite setting temperature (140-160°C).

The most popular in the family of *wrought alloys* (Table 10) is duralumin, which is, in fact, an Al — Cu — Mg alloy. Also widely applied are alloys with admixtures of Mn, Si, Fe and Cr.

The duralumin-type alloys (Д1, Д16, В95) are heat-treated to obtain the highest mechanical properties. The heat treatment consists in water-quenching from a temperature of 500-520°C with subsequent ageing at room temperature for 75-100 h (natural ageing) or at 175-150°C for 1-2 h (artificial ageing). Duralumins are most commonly used in production of sheet and rolled shapes.

To prevent corrosion the rolled stock of aluminium alloys is anodized. The process comprises an electrolytic treatment in a bath of a 20% solution of H_2SO_4 at a current density of 1-2 A/dm^2 and tension of 10-12 V. The component serves as an anode and the cathodes are made in the form of lead plates. As a result of the process the component surface is coated with a film of aluminium oxide Al_2O_3, which effectively protects the metal from corrosion and adds better hardness and abrasion-resistance to the surface. To enhance the stability of the coating it is then treated with a hot 10% solution of potassium bichromate ($K_2Cr_2O_7$).

The sheet rolled stock of aluminium alloys is protected by cladding, i.e., applying upon the surface some layers of pure aluminium.

The AK-type alloys are utilized for forging and stamping of parts (connecting rods of high-speed engines, disks of centrifugal and axial compressors, blades of axial compressors, etc.). The AK4 heat-resistant alloy is used for the manufacture of internal combustion engine pistons and cylinder heads of air-cooled engines.

The wrought aluminium alloys possess satisfactory antifriction properties. The Ni-doped alloys are used in production of plain bearing bushes. An ample circulating supply of lubricating oil for such bearings must be assured for their good serviceability. As for shafts, they must possess increased hardness ($45 > Rc$).

(*b*) *Magnesium Alloys*

The magnesium alloys consist of Mg (90% or more) and of alloying elements (Al, Zn, Mn, Ti, etc.) They have low specific weight ($\gamma \approx \approx 1.8$ kgf/dm^3), low elasticity values ($E = 4200$-4500 kgf/mm^2) and low hardness (60-80 BH). The coefficient of linear expansion of these alloys is very large: $\alpha = (27 \text{ to } 30) \cdot 10^{-6} °C^{-1}$, and the heat conductivity varies between 60 and 70 cal/m·h·°C.

The strength of magnesium alloys is lower than that of aluminium ones and rapidly drops with rise in temperature. The magnesium alloys are sensitive to stress concentrations. They are easily machined but precautions must be taken to avoid ignition of chips.

The magnesium alloys are subdivided into castable and wrought (Table 11).

The worst disadvantage of magnesium alloys is their low corrosion resistance, particularly in humid atmosphere. Therefore, all parts made of magnesium alloys should be adequately protected against corrosion.

Generally this is achieved by dichromizing—a process during which a stable anticorrosion film is formed on the metal surface (the film is composed of magnesium chrome salts).

Table 11

Chemical Composition and Mechanical Properties of Basic Magnesium Alloys

Alloys	Alloy grade	Chemical composition, %			Mechanical properties			
		Al	Zn	Mn	Ultimate tensile strength σ_b, kgf/mm²	Yield limit, $\sigma_{0.2}$, kgf/mm²	Endurance limit, σ_{-1b}, kgf/mm²	Elongation, δ, %
Castable	MJI2	—	—	1-2	8-9	5-6	3-4	2.5-3
	MJI4	5-7	2-3	0.15-0.5	14-16	9-12	6-7	2.5-3
	MJI5	7.5-9	0.2-0.8	0.15-0.5	12-15	9-12	5-7	1.5-2
Wrought	MA1	—	—	1.3-2.5	16-18	10-12	6-9	1.5-2
	MA2	3-4	0.2-0.8	0.15-0.5	24-26	14-18	10-12	3-5
	MA5	7.8-9.2	0.2-0.8	0.15-0.5	28-30	18-20	12-14	6-8

The dichromizing process comprises several stages. First of all the part is treated with a cold 20% solution of chromic anhydride CrO_3 to remove oxide films. Then an electrolytic treatment follows. This is carried out in a bath filled with an acidified aqueous solution of potassium bichromate ($K_2Cr_2O_7$) and ammonium persulphate $(NH_4)_2SO_4$. Finally the surface is treated with a hot 10% solution of chromium anhydride.

Recently selenium treatment has been applied in which the part is treated with a 20% solution of selenium acid (H_2SeO_3) containing a small addition of potassium bichromate.

The part should at least be treated twice: first in the as-cast (as-stamped) condition and then after machining.

Direct contact should be avoided between parts made of magnesium alloys and those made of metals, whose electrochemical potential is higher than that of magnesium (steel, copper alloys, nickel alloys). In this case the parts must be either zinc- or cadmium-plat-

ed. To protect components operating in humid atmosphere (particularly in sea air) the use of zinc or cadmium protectors is recommended.

Magnesium alloys are cast in a protective atmosphere (e.g., in an atmosphere of sulphur dioxide gas produced by powdering the mould interior with sublimed flower of sulphur). However, it is still rather difficult to produce a sound casting with uniform mechanical properties, especially when the casting is large.

Cast magnesium alloys (MJI4, MJI5) are strengthened by means of a proper heat treatment (heating to 380-410°C for 10-18 h, cooling in air and ageing at 175°C for 16-18 h).

In general, magnesium alloys are used for non-power components (non-bearing casings, covers, engine sumps). In particular instances,

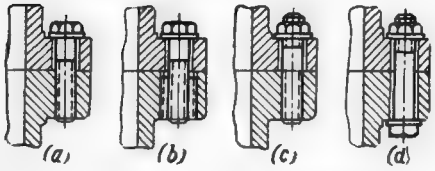


Fig. 96. Connection of light-alloy parts

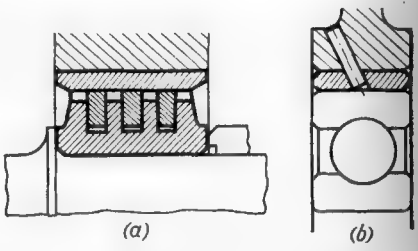


Fig. 97. Reinforcement of light-alloy parts

(*a*) split spring ring seal; (*b*) installation of antifriction bearing in intermediate bush in housing

however, these alloys are used to manufacture some important housings. Wrought magnesium alloys are often used for parts subjected to centrifugal loads.

The disadvantages of magnesium alloys, particularly their poor corrosion-resistance, restrict their application to the cases when eight weight saving is the chief factor.

Particular features of parts made from light alloys. When producing parts from aluminium and magnesium alloys it is necessary to consider their characteristics. Thus, it is possible to compensate for the low strength and rigidity of these alloys by increasing their cross sections and of resisting and inertia moments, by giving the construction a rational form which assures maximum strength and rigidity, and also by suitable ribbing.

The softness and low tensile strength of light alloys forbid the employment of screwed-in fixing bolts (Fig. 96*a*). Should the latter be absolutely necessary for some design reasons, then the holes for tapping must be reinforced with steel bushings (Fig. 96*b*). The best fastenings are studs (Fig. 96*c*) or bolts (Fig. 96*d*) with large steel

washers placed in-between the bolt head and/or nut and the part surface, otherwise, the supporting surfaces are crushed and wear down as the nut is screwed tight.

Friction surfaces in parts made from light alloys should be reinforced with bushes of some hard metal (Fig. 97*a*); antifriction bearings must be mounted in intermediate steel sleeves (Fig. 97*b*).

Light alloy surfaces should never be used for supporting springs, especially when the latter operate under cyclic loads. In such cases

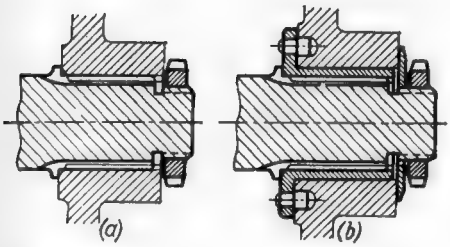


Fig. 98. Spline-fitting of a light-alloy part

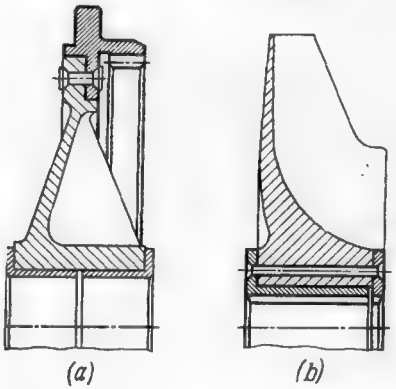


Fig. 99. Composite structures

it is necessary to apply bearing washers, made of some hard metal, so that abrasion of bearing surfaces under the action of multiple repetitive loads is prevented.

Transmission of torque through the agency of keyed or splined connections made directly in a light alloy component is not recommended (Fig. 98*a*). It is better to reinforce the fitting surfaces with steel bushes or transmit torque with the aid of locating screws or pins, spacing them over the radius as much as possible (Fig. 98*b*).

When matching light alloy parts to steel parts make due account for the difference between the linear expansion coefficients of these materials. High thermal stresses can occur in fixed joints, in which the expansion of parts made of light alloys is restricted (locked up) by steel parts. In movable joints, where the male part is made of some light alloy, and its female—from steel (e.g., a cylinder of an internal combustion engine with an aluminium piston) increased clearances should be given so that piston seizure at high temperatures is avoided.

If a part must possess certain qualities (e.g., high hardness, wear resistance, etc.), which light alloys cannot provide, then to make the part lighter use is made of composite constructions. The non-

working portion of a part can be made of light alloy and to this correctly fastened the operative portions which are manufactured from materials having the necessary qualities (Fig. 99). Figure 99*a* shows a composite construction, a cam plate, whose body is made of light alloy whereas the cam rim and the internal drive gear are made of hardened steel. The rim is riveted to the body. Shown in Fig. 99*b*, is an aluminium alloy impeller of a centrifugal compressor; the impeller is reinforced with a steel bush, which has internal driving splines.

(*c*) *Titanium Alloys*

Titanium alloyed with Al, Cr, Mn, Mo, Fe and Si is often used in general engineering. The average specific weight of these alloys is about 4.5 kgf/dm^3, coefficient of linear expansion $\alpha = 8.5 \cdot 10^{-6} °C^{-1}$, thermal conductivity, ≈ 7 cal/m·h·°C.

The main advantages of titanium alloys are: combination of high strength with low specific weight; high heat and corrosion resistance. The strength of titanium alloys rivals that of alloy steels, and their

Table 12

Mechanical Properties of Titanium Alloys

Alloy grade	Ultimate strength, kgf/mm^2			Elongation per unit length, %	Brinnel hardness
	tensile, σ_b	yield, $\sigma_{0.2}$	fatigue, σ_{-1b}		
BT-3	95-115	85-105	40-55	10-16	285-320
BT-4	80-90	70-80	35-40	15-20	285
BT-5	80-95	70-85	35-45	12-25	285-340
BT-6	90-100	80-90	40-50	8-13	320-360
BT-8	105-118	95-110	45-55	6-12	320-380

anticorrosion qualities are better than those of stainless steels. The titanium alloys (Table 12) maintain their high strength within a broad range of temperatures (from minus 200 to plus 600°C). They possess excellent punching and forging characteristics. These alloys are more difficult to machine than steels and require more powerful tools for the purpose. It should be emphasized that under high-speed cutting conditions titanium chips can ignite and the dust is explosive.

To cast titanium is difficult because of the high chemical reactivity of this metal, and for it easily interacts with the moulding materials, as well as the gases given out during casting.

Many titanium alloys can be resistance- and argon-arc welded. Titanium alloys can be subjected to heat treatment (hardening, tempering), chemical-thermal treatment (case hardening, nitriding) and thermal-mechanical treatment. They can also be improved by cold working.

Antifriction properties of titanium alloys are not high. Titanium parts, operating under high friction conditions are nitrided and hardened to 900-1000, VPH. Wear-resistance of titanium parts can also be enhanced through diffusional saturation with copper, tellurium and selenium.

Titanium alloys are extensively used in the aircraft and rocket industries, when the combination of high strength and low specific weight is important. These alloys are indispensable for the manufacture of parts subjected to high inertia loads, in particular, high-speed rotors, in which the stresses are directly proportional to the material's specific weight.

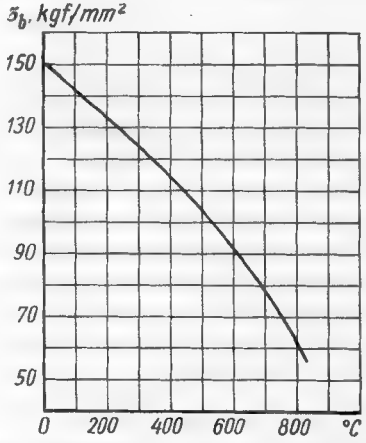


Fig.100. Ultimate tensile strength of titanium alloy grade T12 versus temperature

High heat resistance (Fig. 100) and stability against hot corrosion make titanium alloys suitable for the manufacture of parts working under high temperatures and loads (gas turbine blades). The good corrosion-resistance of titanium ensures its application in the chemical industries.

3.7. Non-Metallic Materials

(*a*) *Plastics*

Plastics (polymers) are, in fact, synthetic high-molecular compounds produced by polymerization or condensation-polymerization of monomers—substances comprising of simple molecules having small molecular weights. Nowadays, we have a wide assortment of plastics, possessing different physical and mechanical properties. The most important features of plastics, when they are used as structural materials, are:

low strength (10-30 times less than that of steels);
low rigidity (20-210 times less than that of steels);
low impact strength (20-50 times less than that of steels);
low hardness (10-100 times less than that of steels);
low heat resistance (100-250°C);
low heat conductivity (100-400 times less than that of steels);
low stability of form, owing to low rigidity, hygroscopicity, creep (inherent in many plastics) and high coefficient of linear expansion (5-20 times that of steel);
low stability of properties; embrittlement when subjected to the prolonged influence of changing temperatures.

Plastics possess excellent dielectric properties and high chemical stability.

Plastics are most commonly applied in electrical engineering, electrical- and radio-instrument industry and chemical engineering. In mechanical engineering plastics are used generally for the production of light-weight housings, covers, panels, controls, decorative elements. Elastic plastics (e.g., PVC, polyolefines, etc.) are widely applied for the manufacture of flexible hoses, sleeves, collars and sealing elements.

Certain plastics (such as, polyamides and fluoroplastics) possess high wear resistance and a low friction coefficient, making them valuable materials for the production of plain bearing sleeves and silent gears.

For load-carrying structures plastics reinforced with fibre-glass and glass-fabric are often applied. Glass-fibre moulding materials are employed for the manufacture of boat hulls, fairings, car bodies and other components of shell-type constructions which successfully

Table 13

Values of C for Metals and Plastics and the Ratio C_{plast}/C_{metal}

Materials	Amino plastics, $C=125$	Vini plastics, $C=170$	Capron, $C=235$	Fibreglass reinforced plastics, $C=680$	Epoxies, $C=1300$	Fluoro-plastics, $C=22\,000$
	C_{plast}/C_{metal}					
Steels:						
carbon ($C=13.5$)	9.3	12.6	17.5	50	95	1600
alloy ($C=12$)	10.5	14	19.5	56	110	1850
stainless ($C=50$)	2.5	3.4	4.7	13.5	26	440
Aluminium alloys ($C=37$)	3.3	4.5	6.5	18	35	600
Bronzes ($C=110$)	1.15	1.5	2.2	6	12	200

compete in strength with similar metallic parts. Insufficient rigidity is compensated for by increased thickness and sections.

It should be noted that plastics constructions for a time remain more expensive than metal ones.

Relative costs of materials are given in terms of specific cost in Table 13, which shows the cost of equal-strength components made from different materials

$$C = \frac{P\gamma}{\sigma_b}$$

where P = price per ton of material in roubles;
γ = specific weight of material, kgf/dm^3;
σ_b = ultimate tensile strength of material, kgf/mm^2

(*b*) *Reinforced Wood*

Wood materials applied in mechanical engineering are generally impregnated with synthetic resins and pressed under high temperatures. The most used are wood-laminated plastics made from the best birch veneer 0.3-1.5 mm thick. The veneer is impregnated with raw bakelite (resol or phenolformaldehyde resin), placed into metal moulds and subjected to a hydraulic pressure of 300-500 kgf/cm^2 at the bakelite setting temperature (160-180°C).

Reinforced wood (*delta-wood*, *laminated birchwood*) has ultimate tensile strength (along layers) σ_b = 15-20 kgf/mm^2, and compressive strength (across layers) σ_{compr} = 25-35 kgf/mm^2 specific weight 1.2-1.4 kgf/dm^3; the mechanical properties, shown for tension across the layers and compression with the layers are 30-40 per cent lower.

Balinite (a wood-resin laminate) is prepared by the same method except that the wood is treated with a 5% NaOH solution prior to impregnation. The mechanical properties of balinite are somewhat higher than those of delta-wood.

Sheets and plates of wood-laminated plastics are widely used for manufacture of panels and various facing pieces. Products from wood-laminated plastics can be moulded into the required shapes.

Lignostone (a kind of laminated wood) is birchwood, bakelite-impregnated and pressed into blocks. This material is used mostly for the manufacture of segment-shaped bearings intended to be operated with water lubrication.

Bushes, gears and other parts are made of birchwood sawdust impregnated with bakelite and then formed under pressure into the required shapes. Wooden gears work well under smooth (impact-free) loads at pressures not exceeding 30-50 kgf per 1 cm of gear tooth width. Wooden gears show high wear resistance when meshed with metal gears.

(*c*) *Glassceramics* (*Sitalls*)

Sitalls (glassceramics) are a silicate glass, possessing a fine-crystalline structure changing radically material properties. They possess improved strength, lack of blittleness and thermofragility inherent in glass and can withstand impact loads successfully.

In contrast to usual glass, which becomes softer with the rise of temperature, sitalls keep their hardness and strength up to 600°C. Like metals, glassceramic materials have a definite melting point, which varies between 1200 and 1400°C depending on the grade.

Their ultimate tensile strength $\sigma_b = 40\text{-}80$ kgf/mm^2, approximates that of carbon steels and high-strength cast irons. In the laboratory sitalls have been obtained with ultimate tensile and compressive strengths of 100 kgf/mm^2 and 150 kgf/mm^2, respectively.

Glassceramics are excellent dielectrics and display high resistance to aggressive chemicals surpassing in this respect plastics, stainless steel and titanium alloys. They also successfully resist the attack of the strongest acids and alkali (except hydrofluoric acid).

Specifications: specific weight 2.2-2.3 kgf/dm^3; thermal capacity 0.2 cal/kg°C; average thermal conductivity 2-4 cal/m·h·°C. Modulus of normal elasticity from 10 000 to 15 000 kgf/mm^2.

An interesting feature of sitalls is possibility of regulating within wide limits their linear expansion coefficient. Depending on the chemical composition and structure of a sitall, its linear expansion coefficient can vary from $20 \cdot 10^{-6}$°C^{-1} to zero. Thus, the possibility is provided of making parts which will not change their linear dimensions despite temperature variations and, hence, will be free of thermal strains. Some sitalls have even negative coefficients of linear extension ($\alpha = -2 \cdot 10^{-6}$°C^{-1}), i.e., their sizes reduce with the rise of temperature.

Sitalls with low linear expansion coefficients are noted for their high thermomechanical stability (products of such sitalls even when heated up to 800-900°C can safely be immersed into cold water). This feature makes sitalls particularly valuable for the manufacture of parts subjected to thermal shocks.

Outwardly, sitalls are glass materials, whose colouring, depending on the chemical composition and structure, can be white, cream, grey, yellow-brownish, brown and dark, down to black. Some sitalls are transparent, others—semitransparent with yellowish or brownish hues.

The main advantage of the glassceramic material is its cheapness and practically inexhaustable raw material reserves. Sitalls are produced from rock minerals: magnesium-alumosilicates, calcium-alumosilicates, calcium-magnesium-alumosilicates (*petrositalls*) or metallurgical slags and cinder (*slagsitalls*).

The process of manufacturing products from sitalls is as follows.

From a charge of the required composition a glass is made from which in the liquid or plastic state the product is formed either by casting or extruding. The obtained products are heat-treated in steps (first step, at 500-700°C and the second, at 900-1100°C) after which they acquire a crystalline structure.

Nucleators are introduced into the glass composition which are substances forming centres of crystallization. Earlier, colloidal particles of Cu, Ag and Au were used as nucleators, which became actually nuclei (seeds) of crystallization as a result of product irradiation by penetrating radiation (*photocerams*).

Nowadays the expensive photochemical process is excluded; as the nucleators now used are iron sulfides, titanium oxides, fluorides and phosphides of alkali and alkali-earth metals (Na, Ca, Li).

At the last step of heat treatment products are uniformly crystallized. The content of a crystalline phase reaches 95% and crystal sizes are rather fine (up to 0.05 μm), i.e., hundreds of times less than those in fine-grained steels. Dimensional changes of a product in the course of crystallization does not exceed 2%.

Crystallized products can be machined by carbide, boron and diamond tools, as well as by ultrasonic technique.

The combination of high strength, toughness, hardness, thermal and chemical resistance, low specific weight and good formability with the use of the most efficient forming techniques make sitalls a promising construction material.

From sitalls are prepared parts for chemical equipment, pumps, heat-exchangers, pipes, vessels, reservoirs, male and female parts of dies, draw plates, parts for radio receivers, electrical machines and instruments.

In civil and industrial construction work sitalls are extensively used as a facing material possessing high strength, durability, abrasion resistance, good heat-insulating properties and complete moisture resistance; furthermore, they resist well the influence of high temperatures, thermal shocks and gas erosion.

Sitalls are also used in the manufacture of thermally stressed parts. Sitall slide bearings can run without lubricant under moderate loads and rotational speeds at temperatures up to 500°C.

Many structural components in general machines may be made of sitalls.

(*d*) *Reinforced Concrete*

In some branches of machine building the use of reinforced concrete structures is promising. It is good practice to prepare from reinforced concrete large-sized housings and basic parts of heavy machines (e.g., beds of unique metal-cutting machines, presses, hammer anvils, etc.). In this case the metal volume is sharply reduced with large savings in manufacturing costs.

For reinforced concrete structures only first-class portland-cement is used, which is a finely powdered silicate mixture roasted before at 1500°C. The mixture is prepared from limestone, clay and quartz sand. Generally, the composition of roasted cement includes: 65-70% CaO; 20-25% SiO_2; 8-10% Al_2O_3 and 2-5% Fe_2O_3. During interaction with water the cement hardens and after a certain interval of time turns into strong monolithic mass. Best hardening conditions require a temperature not lower than 15-20°C and high humidity of the ambient air. Hardening is slower at low temperatures and discontinues at minus temperatures; it accelerates when subjected to heat and moisture treatment (wet steam heating).

The quality of portland-cement is dependent on its mineralogical composition and the fineness of grinding: the finer the cement the more quickly and fully it interacts with water and the higher its strength. Portland-cement usually sets in 1-1.5 h and completely hardens in 10-12 h. With successive maturing the strength of cement increases but after approximately 30 days the hardening process becomes slower.

Portland-cement is produced in the following grades: 200, 250, 300, 400, 500 and 600. The figures indicate ultimate strength in kgf/cm^2 (cube strength), from compression testing a standard cubical sample measuring $20 \cdot 20 \cdot 20$ cm^3, prepared from a mixture of cement and quartz sand (ratio 1 : 3). Testing is carried out after a 28-day hardening period at 15-20°C and 90% relative humidity of the ambient air. The volume weight of portland-cement is 3-3.2 kgf/dm^3.

Concrete is a hardening mass comprising a mixture of cement, fine aggregate (quartz sand) and rough aggregate (gravel). The strength of concrete depends on the quality of cement, as well as on the properties and granulometric composition of aggregates, percentage ratio of cement and aggregates, hardening conditions (ambient temperature and humidity) and also on the method of placement and compaction degree of the mixture.

The weight ratio of concrete constituents is expressed by the following formula:

$$1 : x : y : \frac{W}{C}$$

where 1 = weight of cement, assumed equal to unity;
x = number of parts of sand by weight;
y = number of parts of gravel by weight;
$\frac{W}{C}$ = water-cement modulus, i.e., water-cement ratio

The lower the water-cement modulus, the stronger the concrete. For normal hydration it is enough to introduce water in quantities amounting to 20% of the cement weight $\left(\frac{W}{C} = 0.2\right)$. However, in practice, this ratio is generally taken as equal to 0.3-0.5, because

reduced water content impairs "liveliness" of the concrete mixture. The usual concrete mix is: 1 : 1 : 2 : 0.5.

To produce a strong concrete use is made of a quartz or granite sand, whose particles measure within 0.2-0.4 mm; chippings of approximate size 20-30 mm from hard crystalline rocks (granite, syenite, diabase, basalt). Thin-walled concrete products, walls 30-40 mm thick, are made of cement-sand or cement-chippings mixtures. The chipping size in this case should not exceed 0.25 of the wall thickness.

The strength of concretes is taken as equal to the ultimate compressive strength of the standard cubical sample. As a rule, this cube strength reaches 500-600 kgf/cm^2. When steel chips are utilized as reinforcement (steel reinforced concrete) the cube strength can be as high as 1000 kgf/cm^2.

The volumetric weight of concrete depends on its composition and aggregates. Concrete of the above compositions have volume weights which vary within 2.2-2.7 kgf/dm^3.

Light concretes (volume weight < 1.5 kgf/dm^3) are obtained by employing light sedimentary rocks as aggregates (pumica, tuff, shell rock), as well as cinder or metallurgical slags. Although inferior in strength, light concretes have excellent heat- and sound-proof properties.

Honeycomb and foamy concretes with a volume weight of about 0.2 kgf/dm^3 also possess good heat- and sound-proof qualities.

From the viewpoint of a structural material, concretes display brittleness and sharp anisotropy of mechanical properties. Concrete strength in tension is worse than that in compression and it shows propensity for brittle cracking even when subjected to slight tensile stresses. Its ultimate tensile strength is 10-20 times less than its ultimate compressive strength.

Concretes possess yielding properties. Under compressive loads, exceeding 0.3-0.5 of the cube strength, concretes acquire a yield state and their dimensions spontaneously change which means limiting design compressive stresses to rather low values (150-250 kgf/cm^2 for concretes with a cube strength of 500-600 kgf/cm^2).

Another feature typical of concretes is their low elasticity modulus stipulating the poor rigidity of parts. For concretes the elasticity modulus $E = 1500\text{-}4000$ kgf/mm^2 (average value 3000 kgf/mm^2), being approximately three times less than that of cast iron and seven times less than that of steel. The shear elasticity modulus $G = 1400\text{-}1600$ kgf/mm^2.

Concrete has poor resistance to acids, alkali, machine oils and cutting fluids. To protect concrete from the attack of such compounds it is better employ components covered with a sheet metal skin. Concrete resistance against aggressive chemical substances can be effectively improved by introducing silicone-type polymers into the mixture (polymerconcretes).

The low shrinkage when hardening is a positive feature of concrete as a construction material. The linear shrinkage coefficient of concrete averages 0.03%. This assures the dimensional stability of concrete castings and accuracy of relative position of metallic elements imbedded in the concrete mass. It also reduces machining of metallic base parts of a manufactured product. There are concretes with practically no shrinkage (containing admixtures of gypsum, etc.).

Reinforced concrete, i.e., a concrete with imbedded steel rods, network or lattices, is almost exclusively employed for structures undergoing tensile stresses, as well as dynamic and alternating loads.

The concrete coefficient of linear expansion ($\alpha = 12 \cdot 10^{-6} °C^{-1}$) approaches that of steel, thus ensuring good bond between the concrete and reinforcing elements during temperature fluctuations.

Prestressed reinforced concrete is obtained when the reinforcing elements are subjected to tension during forming process (pretension by jacks or by induction heating), imparting to the concretes higher tensile strength. The weight of steel reinforcement is generally 15-30% of the total weight of reinforced concrete.

Preliminary tensile stresses in the steel reinforcement amount to 150-250 kgf/cm^2. Admissible tensile stresses in the prestressed reinforced concrete average 100-150 kgf/cm^2 and admissible compressive stresses from 300 to 500 kgf/cm^2.

Reinforced concrete possesses an extremely high cyclic toughness, namely, approximately double that of grey cast iron. This feature contributes to the high antivibration properties of concrete products.

It is obvious from the above that reinforced concrete, when used as a structural material, will be inferior to metals in relative strength and rigidity. Permitted tensile and compressive stresses in reinforced concrete are approximately three times less than in grey cast iron. To obtain reinforced concrete structures, equal in strength to the cast iron ones, it is necessary to increase accordingly cross sections and resisting moments. A rule, kept in practice, says that the reinforced concrete products cross sections must be three times greater than those of the cast iron counterparts for the same strength to be achieved. Since the elasticity modulus for reinforced concrete is approximately 1/3 that of cast iron, then increasing reinforced concrete products cross section in the same ratio will bring the rigidity of the latter, when under tension-compression loading, up to the level of the cast iron products.

In practice the rigidity of reinforced concrete structures subjected to tensile-compressive stresses is calculated in terms of the reduced cross section

$$F_{red} = F_c + F_r \frac{E_r}{E_c}$$

where F_c and F_r = areas occupied in the calculated cross-section by concrete and reinforcement, respectively
E_c and E_r = Young's modulus for concrete and reinforcement, respectively

Assuming Young's modulus of concrete $E_c = 3000$ kgf/mm^2 and that of reinforcement $E_r = 21\,000$ kgf/mm^2, we obtain

$$F_{red} \approx F_c + 7F_r$$

simplifying

$$F_{red} = F_{sec}\left(1 + 7\frac{F_r}{F_{sec}}\right)$$

where F_{sec} is the total cross-sectional area subjected to tensile and compressive stresses.

Similarly, when calculating reinforced concrete components subjected to bending, use is made of the reduced resisting moment

$$W_{red} = W_c + W_r\frac{E_r}{E_c} \approx W_c + 7W_r$$

where W_c and W_r are the resisting moments across cross-sections occupied respectively by concrete and reinforcement relative to the neutral axis of the cross-section.

Or

$$W_{red} = W_{sec}\left(1 + 7\frac{W_r}{W_{sec}}\right)$$

where W_{sec} is the moment of inertia across the entire section of part.

In respect of the weight reinforced concrete constructions give way to cast iron ones (of the same strength). The specific weight of reinforced concrete varies within 3-4 kgf/dm^3 depending on the weight of reinforcement. For the increase in the cross-section be only thrice in comparison with the equistrong cast iron part (cast iron specific weight 7.2 kgf/dm^3), the resultant weight of the reinforced concrete components will be $\frac{(3\text{ to }4)\,3}{7.2} = 1.3\text{-}1.7$ times greater than that of cast iron parts.

Thus, the chief gain obtained from the employment of reinforced concrete components is due to decreased metal volume (3-4 times on average) and cheaper production cost. Additional advantages are obtained from simpler production techniques as requirements for pattern-making, moulding and heat treatment of castings are obviated. With correct management of the pouring and hardening proceses waste is practically eliminated.

However, the manufacturing process of machine engineering reinforced concrete components means more manual labour (assembly of moulds, particularly those of metal, positioning and alignment of metallic part locations, placement and tensioning of reinforcement). Another shortcoming is the long production cycle and the necessity to keep the reinforced concrete castings under controlled conditions of temperature and humidity for 15-20 days. The latter disadvantage is eliminated by wet-heat treatment, after which the reinforced concrete attains 70 % of its designed strength within 6-8 h.

Use of reinforced concrete is justified in the manufacture of specially designed and individually produced large-size machines and assemblies. Casting base parts for such machines from cast iron presents formidable difficulties. In some cases (when lacking sufficiently powerful casting machines) the employment of reinforced concrete structures is the only practical solution.

In general machine building, as a means of increasing strength and rigidity, the concrete material can be used to fill hollow spaces in hollow parts. In other instances it is advisable to fill tubular or box-shaped components (uprights, columns, pedestal legs, brackets, beams, etc.) with concrete.

Machine building structures of reinforced concrete. Reinforced structures applied in machine building are essentially frame-reinforced castings, into which steel or cast iron elements are imbedded (pilot bushes, inserts, locating plates, pedestals, uprights, brackets, etc.) to suit the expected purpose of a structure.

There are two principal manufacturing techniques.

The first method: reinforced concrete components are cast in wooden frames which are removed after the concrete has set. The second method: concrete components are cast in welded thin-sheet skins (moulds) 1.5-2 mm thick, internally reinforced with longitudinal and transverse braces (named *permanent metallic framings*). To avoid bulging from the hydrostatic action of the liquid concrete when pouring the shell is additionally strengthened on the outside with removable wooden sections. Holes provided in the shells for risers and gates are welded after the casting has set.

The second method is more perfect, as the metal shell isolates the concrete from the influence of external medii (namely, lubricating oils, lubricating and cooling fluids) and protects it from chance mechanical damage, spalling and shear. However, this method is much more expensive and requires more labour than the first one.

Of utmost importance for the casting strength is the uniformity and compactness of the mould filling. It is obligatory to vibrate the mould at a frequency of 1000-3000 oscillations per minute for 5-10 minutes when filling.

The best results are obtained when the mould is installed and shaken in a vibrostand. When manufacturing large-size components

consolidation is effected from the surface with the aid of plate and rake vibrators. When compacting difficult-to-reach areas depth vibrators are used.

If casting is performed in wooden or open metallic moulds, the humidity of ambient air in the shop should be maintained within 80-90%. To avoid too quick drying open sections should be moistened. After the casting has been freed from its mould (usually in 10-12 days) it must be covered with damp sawdust. These requirements are less rigorous when casting is done in metallic moulds as sufficient moisture is inside the mould.

The density of finished castings is checked by X-ray and ultrasonic techniques.

After seasoning the casting for 15-20 days, its metallic bearing surfaces can be machined. This interval can be cut down to 1-2 days if the casting has been subjected to steaming.

Design rules. When designing reinforced concrete castings the following rules should be observed:

simplify in every possible way the shape of castings, particularly when pouring into metallic shells (moulds), make casting elements in the form of simple geometrical bodies (cylinders, tubes, cones, prisms, etc.);

make wall thicknesses not less than 30-40 mm;

assure smooth transitions from section to section; avoid, whenever possible, difficult-to-fill cavities and pockets in which pits and voids may form; to fill such spaces properly, provide additional risers and gates;

for complex configurations use a livelier concrete with an increased water-cement ratio (0.7-0.8);

in the direction of tensioning forces arrange the load-carrying metallic reinforcement; in zones subjected to bending concentrate the reinforcement at the places where maximum tensile stresses act.

Large-size irregularly shaped components are better made in separate parts. At butt jointing surfaces of parts connected by bolting or welding, suitable metallic fasteners should be cast-imbedded into the concrete. It should be taken into account, however, that jointing lessens the rigidity of the construction, impairs the accuracy of relative position of basic structural surfaces, furthermore, also means more machining.

Constructions, as a whole, must possess a rigidity sufficient not only for normal functioning in stationary conditions but also for transportation and installation in its place. Extensive constructions must include sturdy longitudinal reinforcement elements made of high-grade rolled stock.

Basic metallic components, being cast into the concrete, must be reliably secured and relieved of stresses as much as possible. Such components should not be employed as stiffening elements. Stiff-

ness must be assured by internal reinforcement and suitably shaped sections.

The adhesion strength of the metallic shell and concrete can be enhanced by welding wire or plate anchors to the inside of facing sheets.

3.8. Specific Indices of Strength of Materials

The weight advantages of materials can be evaluated by specific indices characteristic of every type of loading.

Tension-compression. The weight of parts, subjected to tension or compression, other conditions being equal (identical lengths and loads) is calculated by the formula

$$G = \text{const}\, F\gamma \tag{3.22}$$

where F = cross-sectional area of the part;
γ = specific weight of the material

The cross-sectional area is inversely proportional to the acting stress

$$F = \frac{\text{const}}{\sigma}$$

For parts of equal strength (i.e., parts with equal safety margins) the stress value σ can be changed by the ultimate strength value σ_b. In this case

$$F = \frac{\text{const}}{\sigma_b}$$

Inserting this expression into Eq. (3.22), we obtain

$$G = \text{const}\, \frac{\gamma}{\sigma_b}$$

Consequently, the weight of equistrong parts subjected to tensile or compressive stresses under other equal conditions is determined by the ratio between the material's ultimate strength and specific weight

$$\frac{\sigma_b}{\gamma} \tag{3.23}$$

This factor, termed specific strength, characterizes the weight advantage of the material under tension or compression.

Modern design practice diverts from the evaluation of strength in terms of σ_b, i.e., according to failure stresses. Much more often the design calculations pertaining to statically loaded parts are based either on the proportionality limit σ_p (when this value is sharply expressed in the material) or on the yield limit $\sigma_{0.2}$ (i.e., the stress at which the residual strain comprises 0.2 %).

When dealing with parts operating under cyclic loads the initial value for the calculation is the fatigue limit σ_D.

The yield limit is not proportional to the ultimate strength. The $\sigma_{0.2}$ value for different materials varies between 0.5 and $0.95\sigma_b$. Therefore any calculations based on the ultimate strength, even with large safety margins, may lead to gross errors.

As the ultimate strength belongs to the most readily definable characteristics and since a certain proportionality exists between σ_{02} and σ_b for some groups of materials, the σ_b parameter is often taken as the basic strength characteristic.

When calculating in terms of the yield limit $\sigma_{0.2}$, the specific strength factor will be expressed as

$$\frac{\sigma_{0.2}}{\gamma} \tag{3.24}$$

and when calculating in terms of the fatigue limit

$$\frac{\sigma_D}{\gamma} \tag{3.25}$$

This factor is called the specific fatigue limit when under tension or compression.

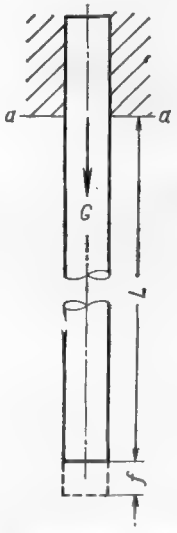


Fig. 101. Freely hanging bar

The specific strength indices can be illustrated by the following.

Imagine a freely hanging bar of an arbitrary section. One end of the bar is secured (Fig. 101) and the bar is, thus, loaded with its own weight. The most dangerous section is *a-a*, upon which acts the full weight

$$G = FL\gamma \tag{3.26}$$

where F = area of cross-section;
L = length of the bar;
γ = specific weight of the bar material

The tensile stress across this section will be

$$\sigma = \frac{G}{F}$$

or, taking into account Eq. (3.26),

$$\sigma = L\gamma \tag{3.27}$$

Let the tensile stress attain the ultimate strength value ($\sigma = \sigma_b$). This will occur when the bar has a certain length L_{br} (i.e., breaking length) which, according to Eq. (3.27), is

$$L_{br} = \frac{\sigma_b}{\gamma}$$

This value coincides with the specific strength of the material, Eq. (3.23).

Should σ_b be expressed in kgf/mm^2, and γ, in kgf/dm^3, then

$$L_{br} = \frac{\sigma_b}{\gamma} \frac{\text{kgf}}{\text{mm}^2} \cdot \frac{\text{dm}^3}{\text{kgf}} = \frac{\sigma_b}{\gamma} \cdot 10^6 \text{ mm} = \frac{\sigma_b}{\gamma} \text{ km} \qquad (3.28)$$

i.e., in this case the length L_{br} will be expressed in kilometers.

Similarly

$$L_y = \frac{\sigma_{0.2}}{\gamma} \frac{\text{kgf}}{\text{mm}^2} \cdot \frac{\text{dm}^3}{\text{kgf}} \qquad (3.29)$$

expresses the length (in km) of a freely hanging bar, at which the stresses across the critical section reach the yield limit.

The displacement of the bar free end (complete extension) as is well known equals

$$f = \frac{GL}{2EF}$$

Since $G = FL\gamma$ and $L = \frac{\sigma}{\gamma}$, then, when $L = L_y$ and $\sigma = \sigma_{0.2}$

$$f_y = \frac{\sigma_{0.2}^2}{2E\gamma} \text{ km} = 10^3 \frac{\sigma_{0.2}^2}{2E\gamma} \text{ m} \qquad (3.30)$$

where $\sigma_{0.2}$ and E are in kgf/mm^2, and γ, in kgf/dm^3.

The f_y value defines the resilience and impact resistance of the material.

Flexure (bending) and torsion. For bending and torsion the weight advantage of the material is computed by the proportion

$$\frac{\sigma^{2/3}}{\gamma} \qquad (3.31)$$

where σ is the rupture stress for the given type of load (σ_b for flexure and τ_b for torsion).

This factor is known as the specific flexural (or torsional) strength. If calculations are carried out in terms of yield or fatigue limit, then the numerical values of corresponding limits are substituted into the numerator of Eq. (3.31).

The weight advantage evaluation is only an approximation. This is why the comparison of all types of loading is generally accomplished with the aid of such structurally simple factors as $\frac{\sigma_b}{\gamma}$ and $\frac{\sigma_{0.2}}{\gamma}$, pertinent to compressive-tensile loads.

Impact loads. The ability to resist impact loads is defined through the work of recoverable strain. When tensioning a bar of constant cross section F and length L

$$U = \frac{P^2L}{2EF} = \frac{\sigma^2FL}{2E}$$

The value of U at stress σ equal to proportionality limit σ_p shows the material's ability to withstand impact loads within the region obeying Hooke's law

$$U = \frac{\sigma_p^2}{2E}$$

Having divided this value by weight $G = FL\gamma$, we obtain

$$^{\gamma}J = \frac{U}{FL\gamma} = \frac{\sigma_p^2}{2E\gamma} \tag{3.32}$$

This factor is the specific impact strength, which characterizes the material weight advantage under impact loads.

When approximate comparison will suffice, the limit of proportionality σ_p can be changed to the yield limit $\sigma_{0.2}$ (which is rather close to it). Then

$$U = \frac{\sigma_{0.2}^2}{2E\gamma} \tag{3.33}$$

This Equation is similar to Eq. (3.30) which pertains to the complete extension f_y of a freely hanging bar of length L_y, at which the stresses in the critical cross section reach the yield limit.

(*a*) *Comparative Weight Evaluation of Structural Materials*

Table 14 cites average values of σ_b, $\sigma_{0.2}$ and E, as well as specific strength characteristics of structural materials. For certain specific strength indices the upper values of σ_b and $\sigma_{0.2}$ are given.

High-strength steels, whose specific strength, expressed in terms of a breaking length (Fig. 102*a*) reaching 45 km, have the best weight advantage. Next to these are glass-fiber anisotropic materials (GFAM) with $L_{br} = 37$ km (this value is valid only for those cases when fibers are favourably oriented with respect to the loads). The last place is occupied by grey cast irons ($L_{br} = 5$ km).

Approximately the same order is preserved when classifying materials in terms of their specific strength $\frac{\sigma_{0.2}}{\gamma}$ (Fig. 102*b*) and specific impact strength $\frac{\sigma_{0.2}^2}{2E\lambda}$ (Fig. 102*c*). In the latter case the extra-high-strength steels are well ahead of the rest of materials, because their specific impact strength is 2, 5 and 40 times better than that of titanium alloys, alloy and carbon steels, respectively.

It is necessary to say that the choice of a material is governed not only by strength-weight characteristics, but also by the purpose and operational conditions. When selecting a suitable material attention is paid to such characteristics as rigidity, hardness, toughness, plas-

Table 14

Specific Strength Characteristics of Structural Materials

Material		Specific weight, γ, kgf/dm³	Ultimate tensile strength, σ_b, kgf/mm²	Yield limit, $\sigma_{0.2}$, kgf/mm²	Modulus of elasticity, E, kgf/mm²	Specific indices		
						$\frac{\sigma_b}{\gamma}$ (L_{br} strength in km)	$\frac{\sigma_{0.2}}{\gamma}$ (L_y strength in km)	$\frac{\sigma_{0.2}^2}{2E\gamma}$ (f_y impact strength in m)
Carbon-steels		7.85	35-80	21-48	21 000	10	6	7
Alloy steels			100-180	80-145		23	18.5	64
High-strength steels			250-350	225-315		45	40	300
Grey cast irons		7.2	20-35	15-25	8 000	5	3.5	5.5
High-strength cast irons		7.4	45-80	32-56	15 000	11	7.7	13
Aluminium alloys	cast	2.8	18-25	13-17.5	72 000	9	6.5	8
	wrought		40-60	28-42		21.5	15	44
Magnesium alloys	cast	1.8	12-20	8-13	4 500	11	7	10.5
	wrought		25-30	16-20		16.5	11	25
Structural bronze		8.8	40-60	32-48	11 000	7	5.5	12
Titanium alloys		4.5	80-150	70-135	12 000	33	30	170
Structural plastics	delta-wood	1.4	15-20 (along layers)	—	5 000	13	—	—
	glass fibre	1.6	25-30	—	5 000	19	—	—
	GFAM	1.9	40-70 (along the lay)	—	6 000	37	—	—
Sitalls		3	50-80	45-72	15 000	27	24	58

ticity, technological characteristics (machinability, punchability, weldability), wear, corrosion and heat resistance, thermal stability (the two latter terms are for parts working under high temperatures).

Of no less importance is the cost of a material, its short supply and expensive and/or difficult-to-obtain components.

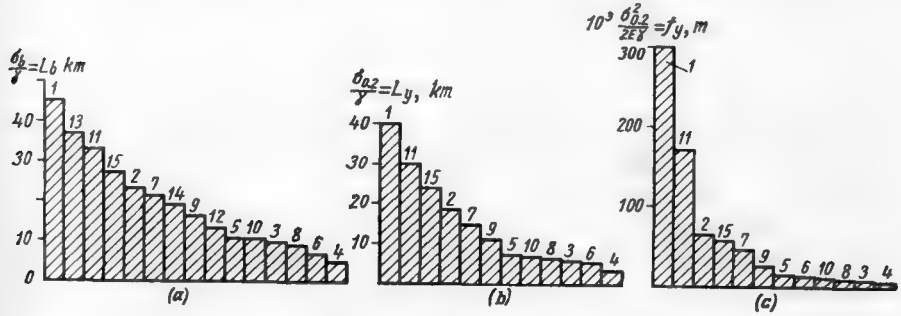


Fig. 102. Specific strength indices of materials

1 — extra-high-strength steels; *2* — alloy steels; *3* — carbon steels; *4* — grey cast irons; *5* — high-strength cast irons; *6* — structural bronzes; *7* — wrought aluminium alloys; *8* — cast aluminium alloys; *9* — wrought magnesium alloys; *10* — cast magnesium alloys; *11* — titanium alloys; *12* — delta-wood; *13* — GFAM; *14* — glass-fibre materials; *15* — sitalls

The best universal properties with high strength-weight indices belong to alloy steels. By introducing alloying elements and applying special heat-treatment processes, it is possible to change their properties within wide limits adding, depending on the requirements, hardness, heat and corrosion resistance, etc. This makes steel become the most widely used and universal material for the manufacture of load-carrying structures.

The same properties of rigidity and high strength-weight indices are possessed by titanium alloys.

Chapter 4

Rigidity of structures

Rigidity is one of the basic factors, which determine the work-capability of a design, and has the same, if not greater, effect on reliability as the strength.

Increased strains can disturb normal functioning of a construction long before stresses dangerous for strength appear. Disturbing the uniform load distribution, they cause a concentration of forces at separate parts of the part, the result being local stress concentrations far exceeding the nominal stress values.

Non-rigidity of housings spoils the arrangements of mechanisms installed inside and causes increased friction and wear of movable joints; non-rigidity of shafts and gear-carrying supports disrupts gear engagement leading to quicker gear tooth wear; non-rigidity of trunnions and sliding bearing supports causes higher edge pressures, appearance of local semi-liquid and semi-dry friction spots, overheating, seizure or shorter bearing life; non-rigidity of fixed joints subjected to dynamic loads brings about fretting, galling and sticking.

In machines performing accurate operations, for example, metal-cutting machine tools, the rigidity of operative members as well as their supports determine the dimensional accuracy of the finished workpieces.

Rigidity parameter is of the greatest importance for the lighter-class machines (transport vehicles, aircraft and rockets). In pursuit of lighter constructions and maximum utilization of materials' strength reserves the designer in given instance is compelled to raise stress levels which leads to larger strains. Extensive employment of equally strong constructions with the most favourable use of weight in its turn causes greater strains as the equistrong structures possess less rigidity.

Particular attention is given to rigidity problems especially with the appearance of high- and super-high tensile strength materials whose application sharply increases the deformation of constructions.

Not uncommon are the cases when the magnitude of forces, affecting the design, is underestimated. Very often during calculations very low values of working forces are obtained while the actual loads unexpectedly reach very high levels leading to partial or complete

failure of a part. This may be caused by imperfect assembly, flexure of insufficiently rigid constructional elements, residual strains, overtightening of fasteners, higher friction, distortion and seizure of rubbing parts, loads occurring during transportation and installation of a machine and other factors not considered at the design stage.

Strain values can be calculated only for the simplest cases, i.e., for ones, which can be solved through routine solutions based on the strength of materials and elasticity theories. However, in practice, the designer has to deal with "incalculable" parts for which it is actually impossible to even approximate the amount of future strains.

Under such circumstances the designer has to recourse to simulating techniques, experiments, including data on practical use of similarly designed machines in the field and sometimes depend on his experience gained over many years. An experienced designer knowing the direction and magnitude of acting forces evaluates more or less correctly their direction and value, amount of future deformation, finds the weakest links and, applying various techniques, increases rigidity, thus creating a rational design.

Conversely, constructions designed by beginners usually have insufficient rigidity.

4.1. Rigidity Criteria

Rigidity is the capability of a system to resist the action of external loads with the least deformation. In mechanical engineering rigidity can be defined thus: rigidity is the capability of a system to resist the action of external loads with permitted deformations which do not destroy the work-capability of the system. The inverse of rigidity is elasticity, i.e., the property of a system to acquire comparatively large deformations under the action of externally applied loads. In respect to machine-building the most important factor is rigidity, however, in certain cases elasticity does play the major role (springs, shock absorbers and other elastic parts).

The rigidity characteristic is evaluated by the coefficient of rigidity (or stiffness), which is the ratio between the force P applied to a system and the maximum deformation f produced by the force.

In the simplest case of tension-compression of a beam of constant cross-section within elastic limits the coefficient of rigidity, in accordance with Hooke's law, will be

$$\lambda_{tens} = \frac{P}{f} = \frac{EF}{l} \tag{4.1}$$

where E = elasticity modulus of material;
F = cross-section of beam;
l = beam length measured along action of force

The inverse value

$$\mu = \frac{f}{P} = \frac{l}{EF} \tag{4.2}$$

characterizing the elastic yielding capacity of the beam is called the coefficient of elasticity.

Determined according to relative strain $e = f/l$, the coefficient of rigidity

$$\lambda'_{tens} = EF \text{ kgf}$$

indicates, in fact, the load in kgf causing relative strain $e = 1$.

The corresponding coefficient of elasticity

$$\mu' = \frac{1}{EF} \text{ kgf}^{-1}$$

gives the relative strain under application of 1-kgf load.

For a beam of constant cross section subjected to torsion, the coefficient of rigidity is the ratio between the applied torque moment M_{trqe} and the angle φ through which the beam sections twist over the length l

$$\lambda_{tors} = \frac{M_{trqe}}{\varphi} = \frac{GI_p}{l} \tag{4.3}$$

where G = the shear modulus of material;
I_p = polar moment of inertia of beam section

The coefficient of rigidity for a beam of constant cross-section upon which a flexural load acts, is given by

$$\lambda_{flex} = \frac{P}{f} = a\,\frac{EI}{l^3} \tag{4.4}$$

where I = moment of inertia of beam section
l = length of beam
a = coefficient depending upon loading conditions

Figure 103 illustrates values of the rigidity coefficient for some forms of flexural loads. Taken as unity is the λ value which corresponds to the flexure of a simply supported beam loaded by a concentrated force P at its midspan.

It is clear that the rigidity of the system depends on the loading conditions. Thus, a beam having a uniformly distributed load (Fig. 103*b*) has a rigidity 1.52 times greater than that of a beam loaded with a concentrated force of the same total value (Fig. 103*a*). The rigidity is affected still more by the type and arrangement of supports. For instance, the rigidity of a simply supported beam with encastre ends (Fig. 103*c*, *d*) is 5-8 times greater than that of a beam freely supported at its ends (Fig. 103*a*, *b*).

The rigidity of a cantilevered beam loaded with a concentrated load (Fig. 103*e*) is only 0.063 of the rigidity of a simply supported beam of the same length loaded with the same force at the midspan.

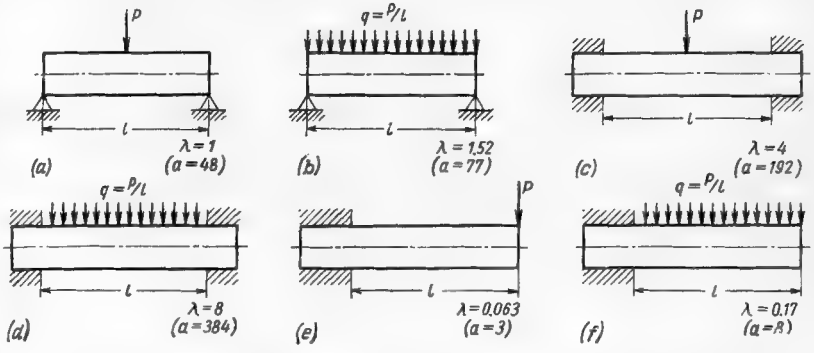


Fig. 103. Rigidity coefficient γ for different flexure schemes

For given load values and linear dimensions of a system the rigidity is determined by the maximum deformation value f. The latter is often employed for the practical evaluation of deformation in geometrically similar systems.

(*a*) *Factors Determining the Rigidity of Constructions*

The rigidity of constructions is governed by the following factors:

modulus of elasticity of the material (modulus of normal elasticity E under tension-compression in bending and shear modulus G in torsion);

the cross-section's geometrical characteristics of the deformed body (section F for tension-compression and moment of inertia I in flexure, the polar moment of inertia I_p in torsion);

linear dimensions of the body undergoing deformation (length l);

type of loading and type of supports [factor a in Eq. (4.4)].

The modulus of elasticity is an extremely stable characteristic of metals and depends solely on the density (packing) of the atomic-crystalline lattice, i.e., on the magnitude of the mean interatomic distance. From all the commercial metals only three possess high moduli of elasticity, namely: tungsten, molybdenum and beryllium (E = 40 000, 35 000 and 31 000 kgf/mm^2, respectively).

The modulus of elasticity of practically all structural materials being used varies from 22 000 (steel) to 4500 kgf/mm^2 (magnesium alloys). However, actual application of a material will mostly depend on operating conditions for the part. Therefore the main practical

means of improving rigidity is by maneuvering the geometrical parameters of the system.

The dimensional and sectional characteristics of a part also have a strong bearing on rigidity. In the case of tensile-compressive loads rigidity is proportional to the square of the cross-sectional sizes, and for flexural loads, to the fourth power (in the direction of the acting bending moment).

The effect of linear dimensions of a part is rather moderate for tension-compression cases (rigidity is inversely proportional to the first power of length), but very significant in flexure (rigidity is inversely proportional to the third power of length).

The design factors influencing the rigidity can be combined in one factor (for a constant acting force P)

in tension-compression

$$\lambda_t = \frac{F}{l} \tag{4.5}$$

in flexure

$$\lambda'_f = \frac{I}{l^3} \tag{4.6}$$

For a round beam:

in tension-compression

$$\lambda'_t = 0.785 \frac{d^2}{l} \tag{4.7}$$

in flexure

$$\lambda'_f = 6.25 \cdot 10^{-4} \frac{d^4}{l^3} \tag{4.8}$$

Equal rigidity conditions for beams with different l and d values and loaded with an identical force P are

in tension-compression

$$\frac{d^2}{l} = \text{const}$$

in flexure

$$\frac{d^4}{l^3} = \text{const}$$

The strength of the material indirectly influences the rigidity of the construction.

The maximum deformation of a part can be expressed in the following way:

in tension-compression

$$f = \frac{Pl}{EF} = \frac{\sigma}{E} l \tag{4.9}$$

where σ is the tensile-compressive stress acting in the part;

in flexure

$$f = \frac{Pl^3}{aEI} = \frac{M_{flex}l^2}{aEW} = \frac{\sigma_{flex}l^2}{abE} \tag{4.10}$$

where σ_{flex} = maximum flexural stress acting in the system;
a = coefficient accounting for loading conditions;
$b = \frac{I}{W}$ is a ratio constant for the given form of section

Thus, it is quite obvious that, other conditions being equal, deformations are proportional to stresses. In practice the magnitude of the stresses are generally assumed to be proportional to the strength characteristics of the material; the stresses are, in effect, the ratio between the ultimate stress (or ultimate yield) and the safety factor coefficient. Hence, the higher the strength of the material, the greater the values of stresses admitted and, other conditions being equal, the larger the deformation of the system. Conversely, the smaller the safety factor and the closer the stresses acting in the system to the ultimate strength, the larger the deformation and the less the rigidity of the system.

The simplest method of reducing deformation is to minimize stresses. However, this way is not rational as it entails penalty in weight. In the case of flexure the rational way to decrease deformation is to select the best section shapes, loading conditions, types and positioning of supports. As the influence of the linear parameters in the system during flexure is rather great [Eq. (4.8)], then for the given case the designer uses a number of different ways to enhance rigidity which allow deformation to be lowered tens of times as compared with the original design and sometimes to eliminate flexure altogether.

In the torsional case the rigidity parameter can be efficiently improved by decreasing the length of the part over the portion subjected to torsion and particularly by increasing the polar moment of inertia.

The improvement of rigidity is much more difficult in the tension-compression case since the shape of the section does not play a large role and the deformation depends only on the sectional area, which is determined by the strength requirements. Here, the only way of increasing rigidity is to shorten the length of the system. If the length cannot be changed, there is no possibility of manoeuvring further.

In accordance with Hooke's law, the deformations of a system subjected to tensile-compressive loads are found by Eq. (4.9)

$$f = \frac{\sigma}{E} l$$

and at given values of stress σ and length l, the deformations depend only on the elasticity modulus.

The deformation value depends not only on the maximum stress acting within the system (across the critical cross-section), but also on the law of stress distribution throughout all other sections, i.e., on the form along the length of the part. Equally strong parts (in which stresses are identical across all sections and equal to the maximum) possess the least rigidity.

(*b*) *Rigidity Beyond Margins of Elastic Strains*

In practice it is necessary to concede the possible occurrence of plastic deformations. Even in systems designed to operate within the margins of elasticity it is not unusual to encounter local plastic

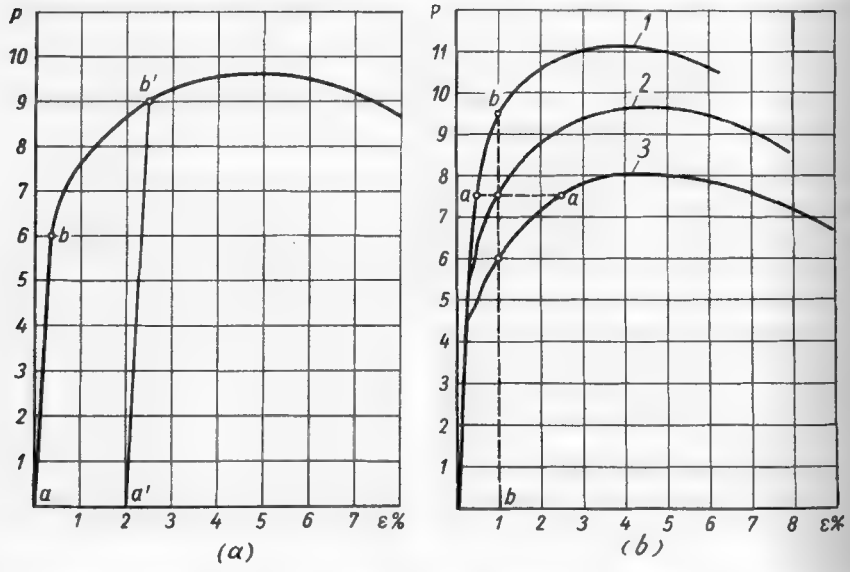


Fig. 104. Loading curve of a plastic metal specimen in tension (*a*) and effect of the strength of steels on plastic deformations of parts (*b*)

deformations in the weakest links of the constructions, at stress concentration spots and in elements unfavourably positioned relative to the acting forces, etc. General or zonal plastic deformations may also occur when operating at overloads.

The behaviour of material in these conditions can be followed from the graph of force and relative extension for the simple tension case (Fig. 104*a*). As long as the part operates within the elastic deformation region (Fig. 104*a*, with loads not exceeding 6 tf), the deformation has an insignificant value ($\varepsilon = 0.2\%$ on average); loading and unloading take place along line *ab*; and when the load is removed, the system each time returns to its original state.

If the acting force rises to a value, surpassing the elastic limit, the deformation is sharply increased due to the appearance of residual strains. For example, with the load increased to 9 tf (point b'), the relative deformation reaches 2.5%. While removing the load, the unloading takes place along line $b'a'$. Even with complete load relief, the system will not return fully to its initial state but will acquire some residual deformation, in this case equal to 2%. At the same time the system attains greater hardness owing to cold working occuring during the plastic flow of the material.

With repeated force applications loading occurs along line $a'b'$ and the system can sustain loads up to 9 tf without any additional residual deformation. However, this lessens the reserves of plastic load carrying capacity (the latter is the difference between the ultimate force and the force corresponding to the elastic limit). If before application of the force which caused the residual deformation the reserves of load carrying capacity amounted to $9.5 - 6 = 3.5$ tf, now it is reduced to $9.5 - 9 = 0.5$ tf.

It is clear that the rigidity drop beyond the elasticity limit is temporary (provided the overloading stress does not exceed the ultimate strength of the material). Having sustained the residual deformation, the system regains its initial state of resilience. The behaviour of the system under repetitive loads is determined by laws of elastic deformation but only with new values of elastic limits and new initial coordinates.

The appearance of moderate residual deformations does not present danger if the load is static and the deformations do not affect the work of the unit and adjacent components. On the contrary, they will help to strengthen the part. The degree of strengthening depends on the ratio between ultimate strength σ_b and ultimate elasticity of the material (or, close to the latter, the yield limit $\sigma_{0.2}$). The $\sigma_{0\ 2}/\sigma_b$ ratio is small for soft and plastic materials (e.g., for low-carbon steels $\sigma_{0\ 2}/\sigma_b = 0.5\text{-}0.6$) and rises with the increase of ultimate strength, reaching as much as 0.85-0.95 in high-strength steels. Thus, the degree of strengthening can be substantial only for plastic materials; the possibility of strengthening high-tensile steels by plastic deformation is not great.

Should residual strains impair the efficiency of a unit (such a case may be encountered in accurate joints), then they must either be eliminated completely, or reduced to very small values.

As evident from the above, the value of the deformation beyond the elastic limits depends primarily on the material's strength and the character of its changes in the plastic deformation area, i.e., on the kind of loading curve.

The illustration in Fig. 104*b* shows comparative values of plastic deformation in parts made from three steels of different strengths. Assume, a part is acted upon by a 7.5-tf tensile force causing stresses

surpassing the elastic limit of all steels. The relative deformation ε under the action of this force (line *aa*) for steels *1* to *3* will respectively be equal to 0.5, 1 and 2.5%. Thereby, the deformation of a part made from the strongest steel is 1/0.5 = 2 times less than that in the case of steel *2* and 2.5/0.5 = 5 times less than that for steel *3*.

The advantages of high-strength steels in the considered case can be illustrated in some other way. Let the given limit relative deformation ε = 1% (line *bb*). The part made from the strongest steel *1* will have this deformation under a 9.5-tf load; from steel *2* — under a 7.5-tf load, and from steel *3* — under a 6-tf load.

Thus, from the above it is clear that the rigidity of a system in the plastic deformation area is determined mostly by the strength factors.

(*c*) *Rigidity of Thin-Walled and Composite Structures*

In thin-walled and particularly shell structures stability of the system is of great importance. Constructions of such a nature are in certain conditions at stresses which are safe from the viewpoint of nominal strength and rigidity calculations prone to abrupt local or general deformations braring the character of collapse.

The main means in the battle against the loss of stability (apart from improvement in material strength) is the reinforcement of easily deformable sections in the system by the introduction of stiffening elements, or braces, between the deforming sections and rigid units.

In composite structures (i.e., in systems, composed of several parts by tight fits) the rigidity will depend also upon a factor, seldom considered, but having great practical significance, namely, the rigidity of matched units. Presence of gaps or clearances in such matched units often results in deformations which surpass many times the natural (inherent) recoverable strains of the structural elements. When dealing with such units special attention must be paid to the rigidity of fastening and built-up parts.

Other efficient ways at enhancing the rigidity of composite systems are dead tightening of joint units, interference fits, larger bearing surfaces and providing greater rigidity at connection points.

4.2. Specific Rigidity Indices of Materials

To compare the indices of rigidity, strength and weight of parts made from different materials, four main cases must be considered:

1. Parts *identical in shapes* (with equal loads the parts have the same stresses).

2. Parts of *equal stiffness* (have equal deformations with different sections and stresses).

3. Parts of *equal strength* (have identical safety factors, different sections and stresses proportional to the ultimate strength of the material).

4. Parts having *identical weight.*

The first case (changing the material of a part for another material but leaving its geometrical dimensions the same) is encountered in practice when the sectional sizes of the part are suited to the manufacturing conditions (e.g., cast housing components). It is the same for parts not specially designed, with small or indeterminate stresses.

The second and the third cases occur when not only the material of a part but also its sectional sizes are changed (designed parts in which stresses and deformations are determined rather closely and specified so as to make the maximum use of the material's strength and rigidity).

The fourth case is when the weight of a structure is determined by its function and conditions of use.

During the comparison of strength, weight and rigidity indices of parts made from different materials, we assume that the parts have equal length and their sections (for the last three cases) are changed in a geometrically similar manner.

1. **Parts of similar shapes** (σ = const). For tension-compression, the rigidity coefficient

$$\lambda = \frac{EF}{l}$$

where F and l = cross-sectional area and length, respectively;
E = Young's modulus

Given: l = const, and F = const, hence

$$\lambda = \text{const}\, E \tag{4.11}$$

i.e., the rigidity of parts in this case is dependent only on the value of Young's modulus.

Safety factor

$$n = \frac{\sigma_b}{\sigma}$$

where σ_b = ultimate tensile strength;
σ = acting stress in a part

Given: σ = const, hence

$$n = \text{const}\, \sigma_b \tag{4.12}$$

The value n defines the maximum load which the part can sustain

$$P_{max} = nP$$

In the case under consideration the maximum load is determined by the metal tensile strength σ_b, and the weight of a part, only by the material's specific weight, i.e.,

$$G = \text{const}\,\gamma \tag{4.13}$$

The correlations are absolutely similar for flexure and torsion, the only difference being that in torsion the rigidity of a part is dependent on the value of the shear modulus.

2. **Equally rigid parts** ($\lambda = \text{const}$). The conditions for equal rigidity in tension and compression

$$\lambda = \frac{EF}{l} = \text{const}$$

With identical lengths l

$$F = \frac{\text{const}}{E} \tag{4.14}$$

Consequently, the weight of equally rigid parts

$$G = F\gamma = \text{const}\,\frac{\gamma}{E} \tag{4.15}$$

Stresses

$$\sigma = \frac{\text{const}}{F}$$

Taking into account Eq. (4.14)

$$\sigma = \text{const}\,E$$

The safety factor (design redundancy)

$$n = \frac{\sigma_b}{\sigma} = \text{const}\,\frac{\sigma_b}{E} \tag{4.16}$$

In flexure the weight of equally rigid parts

$$G = \text{const}\,\frac{\gamma}{E^{1/2}} \tag{4.17}$$

The safety factor (design redundancy)

$$n = \text{const}\,\frac{\sigma_b}{E^{3/4}} \tag{4.18}$$

3. **Equistrong parts** ($n = \text{const}$). Conditions for equal strength in tension and compression

$$n = \frac{\sigma_b}{\sigma} = \text{const}$$

As $\sigma = \dfrac{\text{const}}{F}$

$$n = \text{const}\,\sigma_b F = \text{const}$$

Hence, for equistrong parts

$$F = \frac{\text{const}}{\sigma_b} \tag{4.19}$$

and their weight

$$G = \gamma Fl = \text{const}\,\frac{\gamma}{\sigma_b} \tag{4.20}$$

The coefficient of rigidity is tension-compression

$$\lambda = \frac{EF}{l}$$

Taking into account Eq. (4.19)

$$\lambda = \text{const}\,\frac{E}{\sigma_b} \tag{4.21}$$

In flexure the relations will be

$$G = \text{const}\,\frac{\gamma}{\sigma_b^{2/3}} \tag{4.22}$$

$$\lambda = \text{const}\,\frac{E}{\sigma_b^{4/3}} \tag{4.23}$$

4. **Parts of equal weight** (G = const). The conditions for equal weight in tension-compression

$$G = F\gamma l = \text{const}$$

leads to the relation

$$F = \frac{\text{const}}{\gamma} \tag{4.24}$$

Stresses in a part

$$\sigma = \frac{\text{const}}{F} = \text{const}\,\gamma$$

The safety factor (design redundancy)

$$n = \frac{\sigma_b}{\sigma} = \text{const}\,\frac{\sigma_b}{\gamma} \tag{4.25}$$

The coefficient of rigidity

$$\lambda = \frac{EF}{l} = \text{const}\,\frac{E}{\gamma} \tag{4.26}$$

In flexure

$$n = \frac{\sigma_b}{\gamma^{3/2}} \tag{4.27}$$

$$\lambda = \frac{E}{\gamma^2} \tag{4.28}$$

For the purpose of comparison use is made of the simplest equations pertaining to tension-compression.

For all the above analyzed cases, in tension and compression, the indices of weight, rigidity and strength of parts made from different materials are cited in Table 15. The last two bottom rows give

Table 15

Characteristics of Weight, Rigidity and Strength

Parameter	Parts			
	similar shapes	equal rigidity	equal strength	equal weight
Weight, G	γ	$\frac{\gamma}{E}$	$\frac{\gamma}{\sigma_b}$	const
Rigidity, λ	E	const	$\frac{E}{\sigma_b}$	$\frac{E}{\gamma}$
Strength (n; P_{max})	σ_b	$\frac{\sigma_b}{E}$	const	$\frac{\sigma_b}{\gamma}$
$\frac{\lambda}{G}$	$\frac{E}{\gamma}$			
$\frac{P_{max}}{G}$	$\frac{\sigma_b}{\gamma}$			

relative values of λ/G and P_{max}/G; the first of these indicates the weight advantage in terms of rigidity and the second — in terms of strength.

From the Table it is seen that factor λ/G, of the weight advantage in terms of rigidity, is the same for all cases and equal to E/γ. This is called the specific rigidity for tension-compression; this factor is the basic weight-rigidity characteristic of materials.

The specific rigidity in flexure

$$\frac{E^{1/2}}{\gamma}$$

in torsion

$$\frac{G^{1/2}}{\gamma}$$

where G is the modulus of shear.

In all cases the weight advantage in terms of strength is defined by the specific strength factor (failure length σ_b/γ).

Table 16

Strength and Rigidity Characteristics of Structural Materials

Materials		Specific weight, γ, kgf/dm³	Ultimate tensile strength, σ_b, kgf/mm²	Yield limit, $\sigma_{0.2}$, kgf/mm²	Modulus of elasticity, E, kgf/mm²	Modulus of shear, G, kgf/mm²	Rigidity characteristics			Generalized factor, $\frac{\sigma_{0.2} E}{\gamma} \cdot 10^{-5}$
							$\frac{E}{\gamma} \cdot 10^{-3}$	$\frac{E}{\sigma_b} \cdot 10^{-2}$	$\frac{E}{\sigma_{0.2}} \cdot 10^{-2}$	
Carbon steels		7.85	35-80	21-48	21 000	8 000	2.67	2.6	1.56	1.3
Alloy steels			100-180	80-145				1.17	0.94	3.8
High-strength steels			250-350	225-315				0.6	0.54	8.4
Grey cast irons		7.2	20-35	14-25	8000	4500	1.1	2.3	1.6	0.3
High-strength cast irons		7.4	45-80	32-56	15 000	7000	2	1.9	1.3	1.1
Aluminium alloys	cast	2.8	18-25	13-17.5	7 200	2500	2.67	2.9	2.04	0.45
	wrought		40-60	28-42				1.2	0.83	1.1
Magnesium alloys	cast	1.8	12-20	8-13	4500	1500	2.3	2.1	1.35	0.32
	wrought		25-30	18-21				1.4	1.2	0.52
Structural bronzes		8.8	40-60	32-48	11 000	4200	1.25	1.85	2.3	0.6
Titanium alloys		4.5	80-150	70-135	12 000	4200	2.66	0.8	0.72	3.6
Structural plastics	delta-wood	1.4	15-20 (along layers)	—	4000	—	2.9	2	—	—
	glass-fibres	1.6	25-30	—	5000	—	3.1	1.67	—	—
	GFAM	1.9	40-70 (along fibres)	—	6000	—	3.1	0.86	—	—
Sitalls		3	50-80	45-72	15 000	—	5	1.87	1.68	3.6

The rigidity of equally strong parts is defined by the ratio E/σ_b of the elasticity modulus to the ultimate tensile strength, termed the rigidity factor of equistrong parts under tension-compression.

In flexure the rigidity factor is expressed as

$$\frac{E}{\sigma_b^{4/3}}$$

Table 16 and Fig. 105 describe the specific rigidity characteristics of the principal structural materials, defined in the Table according to the maximum ultimate tensile strength values for the given material.

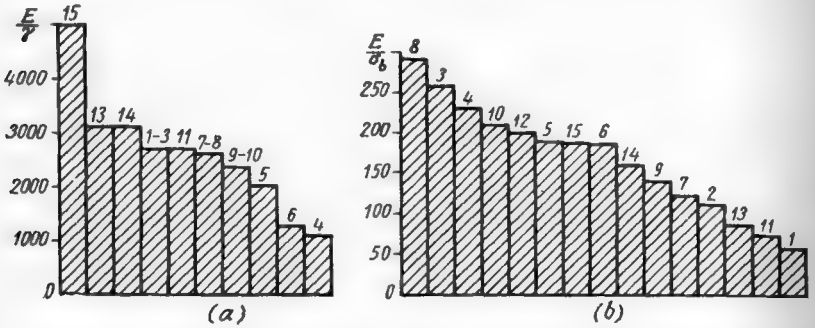


Fig. 105. Specific rigidity factors of materials

1 — superhigh-strength steels; *2* — alloy steels; *3* — carbon steels; *4* — grey cast irons; *5* — high-strength cast irons; *6* — structural bronzes; *7* — wrought aluminium alloys; *8* — cast aluminium alloys; *9* — wrought magnesium alloys; *10* — cast magnesium alloys; *11* — titanium alloys; *12* — delta-wood; *13* — GFAM; *14* — glass-fibre materials; *15* — sitalls

By the magnitude of the $\frac{E}{\gamma}$ factor (Fig. 105*a*) the first place is occupied by sitalls; these are followed by composite plastics, titanium alloys and steels. By the magnitude of the E/σ_b factor (Fig. 105*b*) the strongest materials are found to occupy the last place.

The highest specific rigidity is shown by the beryllium-aluminium alloys (24-38% Al, balance—Be), being applied as yet on a reduced scale. The specific weight of these alloys $\gamma = 2\text{-}2.1$ kgf/dm^3, ultimate tensile strength $\sigma_b = 45\text{-}60$ kgf/mm^2, Young's modulus $E = (20\text{-}22)\cdot 10^3$ kgf/mm^2, relative elongation $\delta = 5\text{-}8\%$. Specific rigidity of these alloys $\frac{E}{\gamma} = \frac{22\cdot 10^3}{2} = 11\cdot 10^3$, i.e., approximately 4 times that of steels. The failure length $L_{fail} = 60 : 2 = 30$ km.

In the case of parts having similar shapes (Fig. 106*a*) the most advantageous are steels for their rigidity (determined by E) and strength (determined by σ_b), but in respect of weight (determined by the specific weight of materials) steels, as well as bronzes and cast irons are worse.

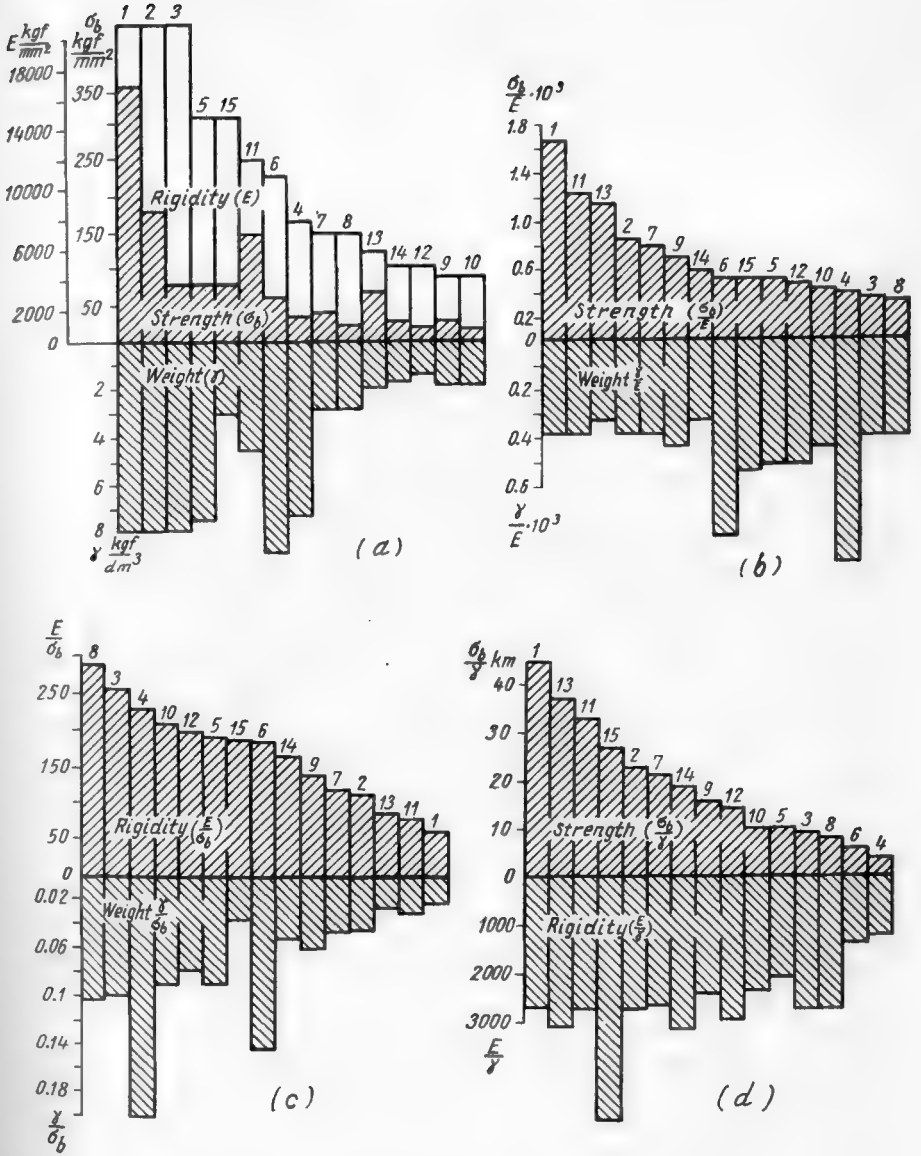


Fig. 106. Influence of material on the weight, strength and rigidity of constructions

(*a*) of similar configurations; (*b*) equirigid; (*c*) equistrong; (*d*) of equal weight; *1* — superhigh-strength steels; *2* — alloy steels; *3* — carbon steels; *4* — grey cast irons; *5* — high-strength cast irons; *6* — structural bronzes; *7* — wrought aluminium alloys; *8* — cast aluminium alloys; *9* — wrought magnesium alloys; *10* — cast magnesium alloys; *11* — titanium alloys; *12* — delta-wood; *13* — GFAM; *14* — glass-fibre materials; *15* — sitalls

Let us compare the rigidity, strength and weight of structures when changing over from castings of pig iron to castings of aluminium and carbon steels without altering the parts' shape. Assuming the rigidity, strength and weight of grey cast iron structures as equal to unity, we obtain:

	Aluminium alloys	*Carbon steel*
Rigidity (E) . . .	0.9	2.6
Strength (σ_b) . .	0.72	2.3
Weight (γ)	0.39	1.1

Thus, the transfer to aluminium alloy castings hardly affects rigidity, somewhat lowers strength (approximately 30%) and significantly reduces the weight of the structure (2.5 times). Changing to steel castings increases rigidity and strength approximately 2.5 times and the weight is practically the same.

The above listed relations are based on the fact that Young's modulus has a constant value and only slightly depends on the presence (in normal amounts) of alloying elements, heat treatment and strength characteristics of alloys of the given metal. For example: in respect to steels, beginning from low carbon and up to high-alloy ones, Young's modulus varies from 19 000 to 22 000 kgf/mm^2, and the shear modulus, from 7900 to 8200 kgf/mm^2. With regard to aluminium alloys E=7000-7500 kgf/mm^2 and G=2400-2700 kgf/mm^2.

Consequently, for the manufacture of identically shaped components when rigidity is the first requirement and the stress level is low, the use of cheapest materials is recommended (low carbon steels instead of alloyed ones; simple aluminium alloys instead of complex ones). This will not affect the rigidity of the construction but will enable the cost of the latter to be reduced.

The above does not hold when the strength of the construction is as important as the rigidity. Thus, for instance, identically shaped structures, one made from low-carbon steel, and the other, from alloy steel, will have the same rigidity but the load capacity of the first structure will be as many times less as the tensile strength of carbon steel is less than that of alloy steel.

For equally rigid parts (Fig. 106*b*) the greatest strength advantage is with materials having the highest σ_b/E ratio (high-strength steels, titanium alloys, GFAM and wrought aluminium alloys). In terms of weight advantage (this is, in our case, proportional to the γ/E factor) the above-listed materials possess approximately equivalent values. Worse weight characteristics are held by bronzes and grey cast irons.

Now, let us compare the strength and weight characteristics of equally rigid parts when changing from carbon steel to wrought aluminium alloys, alloy steel and titanium alloys. Assuming the

strength and weight of the carbon steel structures to be equal to unity, we obtain:

	Aluminium alloys	*Alloy steel*	*Titanium alloys*
Strength $\left(\frac{\sigma_b}{\gamma}\right)$ . . .	2.15	2.3	3.3
Weight $\left(\frac{\gamma}{E}\right)$ . . .	1.05	0.43	1

Hence, for the equally rigid parts, the change from carbon steel to alloy steel, titanium and aluminium alloys is accompanied by a substantial strength increase (from 2.15 to 3.3 times). The construction's weight is lowered 2.3 times when using alloy steels and practically remains the same with aluminium and titanium alloys.

In the case of equally strong parts (Fig. 106*c*) the weight of the structure depends on the magnitude of factor γ/σ_b: in terms of weight the advantages lie with the materials which have a low γ/σ_b factor (sitalls, GFAM, titanium alloys and high-tensile steels). Grey cast irons and bronzes have lower weight characteristics (advantages).

The rigidity of constructions in the considered case depends on the rigidity factor E/σ_b. The greatest rigidity advantages are possessed by materials which have the highest values of this factor, i.e., the less- strong materials (cast aluminium and magnesium alloys, carbon steels, grey cast irons) and the worst rigidity by the high-strength materials (alloy steels, titanium alloys, extra-high tensile steels).

The meaning of these circumstances must be evaluated correctly. All the above is true only in assuming that the design stresses are chosen proportionally to the material's ultimate strength. In this case the use of high-strength materials for equal-strength constructions definitely lowers their rigidity.

Thus, with the same safety factor ($n = 5$) and admissible stresses of 30 kgf/mm^2, the rigidity of alloy steel parts will be one third of the rigidity of equal-strength parts made from carbon steels having admissible stresses of 10 kgf/mm^2. The rigidity of parts made from extra-high-strength steels, admissible stresses 70 kgf/mm^2, will be 2.5 times less than that of the rigidity of equal-strength alloy steel parts and 7 times less than that of equally strong carbon steel parts.

The rigidity of structures made of high-strength materials may be practically unreservedly increased by lowering design stresses, but with detriment to the weight advantage of the structure and incomplete utilization of the materials strength reserves.

Thus, we conclude that] when using high-strength materials in equal-strength constructions it is necessary t oaccount for the fall in rigidity and compensate for this effect by other design measures.

Let us compare the rigidity and weight of equally strong structures when carbon steel is changed by wrought aluminium alloys, alloy steel

and titanium alloys. Having assumed that the rigidity and weight of carbon steel are equal to unity, we obtain:

	Aluminium alloys	*Alloy steel*	*Titanium alloys*
Rigidity $\left(\frac{E}{\sigma_b}\right)$ . . .	0.53	0.51	0.35
Weight $\left(\frac{\gamma}{\sigma_b}\right)$ . . .	0.48	0.45	0.3

Thus, the transfer, in equally strong parts, to aluminium alloys, alloy steel and titanium alloys causes an abrupt (2-3 times) drop of rigidity. Simultaneously, approximately in the same proportion, the weight of constructions is reduced.

With respect to parts of equal weight (Fig. 106*d*) the strength of a structure is determined by the σ_b/γ specific strength factor. Here it is more advantageous to use materials with a high σ_b/γ ratio (high-tensile steels, titanium alloys, GFAM and sitalls). In terms of structural rigidity the above-listed materials are approximately the same, except sitalls which have sharply increased rigidity.

Now, let us compare the strength and rigidity of structures having the same weight, when changing from carbon steel to wrought aluminium alloys, alloy steel and titanium alloys. Assuming the strength and rigidity of carbon steel to be equal to unity, we obtain:

	Aluminium alloys	*Alloy steel*	*Titanium alloys*
Strength $\left(\frac{\sigma_b}{\gamma}\right)$ . . .	2.1	2.25	3.23
Rigidity $\left(\frac{E}{\lambda}\right)$ . . .	1	1	1

Hence, the change to aluminium and titanium alloys and alloy steels leads to increased strength of equal-weight parts by 2-3 times with rigidity unaffected.

(*a*) *Generalized Index*

As shown above the weight advantage of a material in terms of strength is described by the factor σ_b/γ (or $\sigma_{0.2}/\gamma$), and the rigidity advantage is expressed by the factor E/γ. Uniting these we obtain a generalized strength-rigidity index $\frac{\sigma_{0.2}E}{\gamma}$ (See Table 16), which characterizes the capability of a material to sustain the highest loads with the least deformations and least weight of a construction.

Finally, this is very important. Rigidity and strength are inseparable. Rigidity, if taken solely by itself, has no practical value unless the material is capable of sustaining high loads. A slender rod made of soft carbon steel can easily be bent by hands. The rigidity of such

a rod is insignificant and its constructional value is zero. The same rod produced from high-tensile heat-treated steel can carry considerable loads. The rod will be "rigid" even though the elasticity modulus of its material is the same.

Here is an example from mechanical engineering. A hoisting crane manufactured from low-carbon raw steel will have the same rigidity as a crane made from high-grade heat-treated steel, when

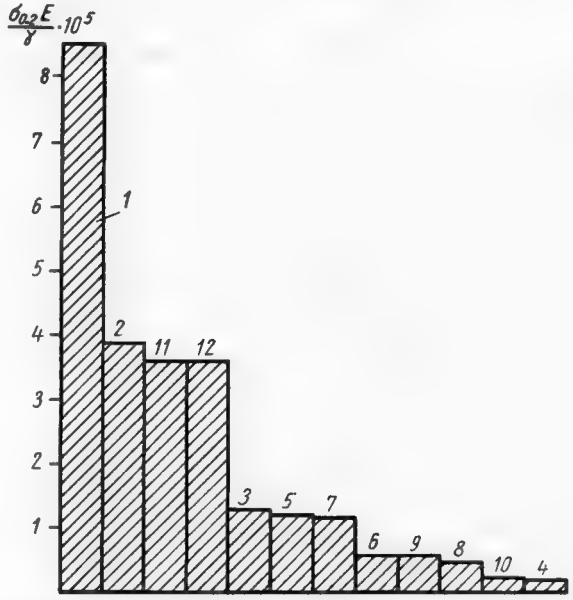


Fig. 107. Generalized strength-rigidity indices

1—superhigh-strength steels; *2*—alloy steels; *3*—carbon steels; *4*—grey cast irons; *5*—high-strength cast irons; *6*—structural bronzes; *7*—wrought aluminium alloys; *8*—cast aluminium alloys; *9*—wrought magnesium alloys; *10*—cast magnesium alloys; *11*—titanium alloys; *12*—sitalls

judged in terms of the material's characteristics. However, the former crane will deform and settle under the action of high loads, which in the latter crane only cause insignificant elastic deformations.

According to the value of the generalized weight and strength-rigidity factor (Fig. 107), the materials are divided into four rather precise groups: (1) extra-high-strength steels $\left(\frac{\sigma_{0.2}E}{\gamma} \approx 8\cdot 10^5\right)$; (2) alloy steels, titanium alloys and sitalls $\left[\frac{\sigma_{0.2}E}{\gamma} = (3.5\text{-}4)\cdot 10^5\right]$; (3) carbon steels, high-strength cast irons and wrought aluminium alloys $\left(\frac{\sigma_{0.2}E}{\gamma} \approx 1\cdot 10^5\right)$; (4) structural bronzes, wrought magnesium alloys,

cast aluminium and magnesium alloys, grey cast irons ($\frac{\sigma_0 2E}{\gamma} < 0.5 \cdot 10^5$).

In practice, the choice of material is determined not only by the strength-rigidity characteristics, but also by some other properties. That is why preference is given to those design measures which enable strength and rigidity to be obtained, even when applying materials of low strength and rigidity.

4.3. Enhancing Rigidity at the Design Stage

The main constructional methods of improving rigidity are:

do all that is possible to eliminate flexure which is unremunerative from the viewpoint of rigidity and strength, changing it to compression and tension;

for parts working in flexure, position supports rationally, exclude loads not advantageous to rigidity;

rationally increase section inertia moments, reinforce joints and the parts transferring forces from one section to another, all without causing weight increases;

for thin-walled box-shaped parts, use shell, arched, vaulted, spherical and oval (egg-shaped) forms.

(*a*) *The Change of Flexure to Tension-Compression*

The higher rigidity of parts, working in tension-compression is due to the better use of the material with this form of loading. In the case of flexure and torsion the most loaded fibres are at the cross-sections extremes. Thus, the loading limit begins when the stresses in these fibres reach dangerous values and the core remains underloaded. In tension-compression conditions the stresses are equally distributed throughout the entire cross-section and the material is fully utilized. The ultimate level of loading arrives only when the stresses in all points of a section simultaneously approach, at least theoretically, some critical level. In addition, the action of loading under tensile-compressive forces is independent of the length of a part; the deformations are proportional to the part length, whereas in the case of flexure the action of load is dependent on the distance between the plane of acting flexural forces and the critical section; deformations in this case are proportional to the third power of the part length.

Let us compare a cantilevered beam of circular cross section (Fig. 108*a*), loaded with a flexural force P, and an equivalent truss system, comprised of bars having the same section as the beam. In the latter case the upper bar under the action of force P works in tension and the lower one in compression. From the relations

presented in the figure, the stresses in the truss bars are 550 times less than the maximum stresses in the beam and the maximum deformation at the application point of force P is 9000 times less.

If the maximum bending stresses in the beam (Fig. 108*b*) equal the tensile-compressive stresses in the bars (equal-strength case of both systems), the cross-sectional area of the beam must be increased

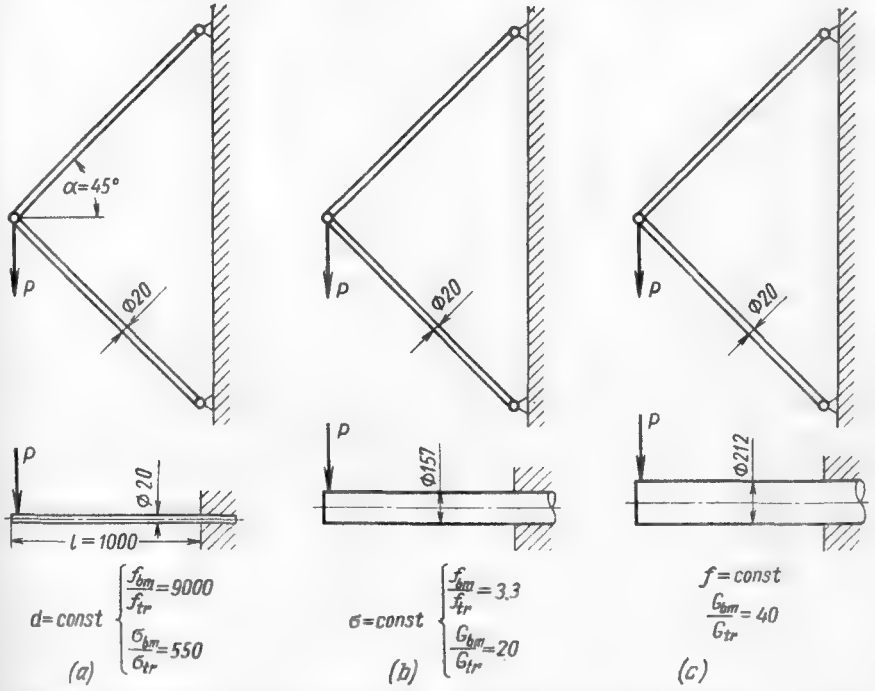


Fig. 108. Rigidity, strength and weight comparison of a truss system and cantilevered beam

(*a*) with equal cross-sections of truss bars and beam; (*b*) with equal stresses; (*c*) with equal deflections

60 times. In this case the weight of the beam, excluding the fixed end, will be 20 times that of the truss system and the flexure of the beam in the plane of the applied load will be 3.3 times that of the truss system.

For equal maximum deformations in both systems (Fig. 108*c*) the beam cross-section must be increased 110 times and its weight will exceed that of the truss system 40 times.

The connection between flexure f_b of the round cantilevered beam (in the plane of applied force) and flexure f_{tr} of the truss system with equal sections (Fig. 108*a*) may be expressed by the approximate

relation

$$\frac{f_b}{f_{tr}} = 10.5 \left(\frac{l}{d}\right)^2 \sin^2 \alpha \cos \alpha$$

where l = span length;
d = beam and truss bars diameter;
α = half-angle taken at the apex of the truss-formed triangle

The relation $\frac{f_b}{f_{tr}}$ as a function of α angle is plotted to a semi-log scale for different values of $\frac{l}{d}$ (Fig. 109). From this graph, with equal

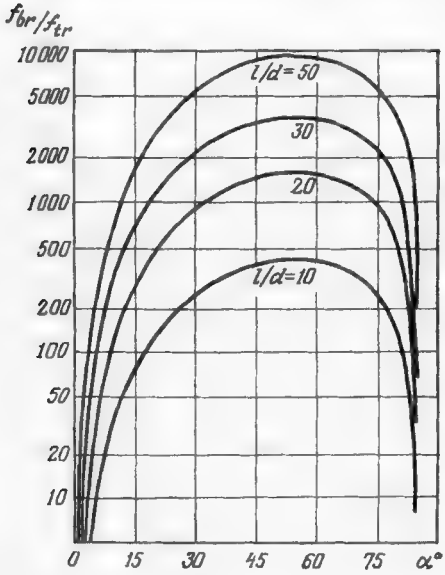


Fig. 109. Ratio between the deflections of a beam and triangular truss system (as a function of angle α)

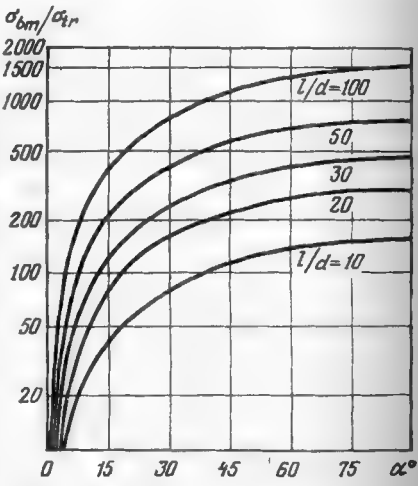


Fig. 110. Ratio between the stresses in a beam and triangular truss system (as a function of angle α)

sections, it is seen that the flexure of the cantilevered beam can be hundreds and thousands times greater than that of the truss system, the difference sharply increasing with the enhancement of the l/d ratio, i.e., with relative thinning of the beam and bars. Even for the most rigid bars ($l/d = 10$) the difference remains in favour of the trussed bar system.

The f_b/f_{tr} ratio reaches its highest value when $\alpha = 45\text{-}60°$. This means that the truss shown in Fig. 108 possesses its maximum rigidity when $\alpha = 45\text{-}60°$.

Having plotted the dependence of the ratio of maximum flexural stress σ_b in the beam to the stresses of tension-compression σ_{tr} in the truss bar system on angle α (for various l/d values) (Fig. 110)

it is possible to establish that the stresses in the beam are many times those in the truss bars (e.g., when $\alpha = 45°$, 100-1000 times).

A cast bracket (Fig. 111*a*, *b*) is a structure similar to the systems depicted in Fig. 108. The rigidity of the bar connecting units at the bracket stays changes their operating conditions as compared to the pure truss in which the bars are articulated; nevertheless, for

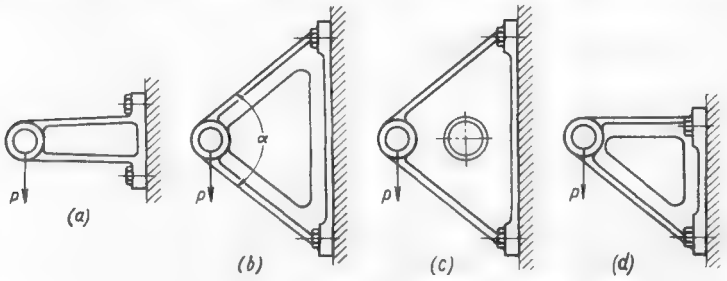


Fig. 111. Designs of cast brackets

the truss-type bracket (Fig. 111*b*) the bars work mainly in tension-compression, but the cantilevered form of bracket (Fig. 111*a*) will be subjected to flexure.

The structure becomes stronger and more rigid if the bars are joined together by a solid web converting the entire unit into completely rigid system (Fig. 111*c*).

The truss-type bracket with the horizontal upper bar shown in Fig. 111*d*, is much less rigid than the bracket shown in Fig. 111*b*, because the end of the horizontal bar, when subjected to loads, is displaced approximately along the direction of the acting force and its rigidity is not used for limiting deformations.

Design calculations show that the best value of angle α, when considering rigidity and weight, lies between 90 and 120°.

In the short thin-walled cylinder carrying transverse load P (Fig. 112*a*) all the sections, positioned along generatrices undergo flexure. The loads are mainly taken up by the side walls (Fig. 112*b*) which are parallel to the bending moment action (blackened in the Figure), since their rigidity in this direction is many times more than the rigidity of the walls positioned perpendicularly to the plane of the acting moment.

In the conical form (Fig. 112*c*) approximating a truss-type construction (Fig. 108) the walls positioned in the plane of the acting bending moment work as follows: the upper wall in tension, the lower wall resembling a strut in compression. The side walls mainly undergo flexure and their rigidity is commensurable with the rigidity of the upper and lower walls. Consequently, all walls of a conically shaped component take part in the work and the strength and rigidity of the construction is increased.

Connection between the tensioned and compressed walls is accomplished by a ring of stiffness m, set at the end of the unit. Apart from acting as a load-closing element, this ring prevents the cone becoming oval under the action of the load. Such rings are indispensable for thin-wall hollow components to operate correctly.

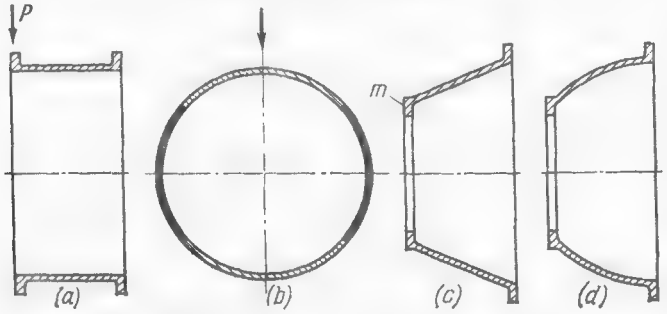


Fig. 112. Cantilevered thin-walled systems

Close to the conical components in terms of rigidity are spherical (Fig. 112*d*) and other convex-shaped parts.

Another example of eliminating flexural stresses is given in Fig. 113. Here, a simply-supported beam, subjected to flexure (Fig. 113*a*) is changed to a much more advantageous bar structure,

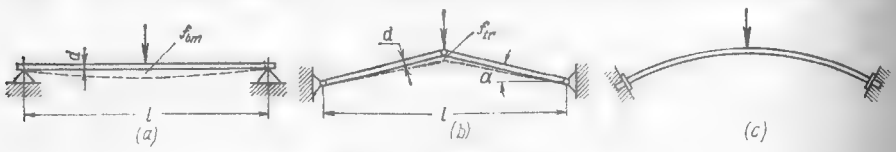


Fig. 113. Schemes

(*a*) freely supported beam; (*b*) truss system; (*c*) arched beam

whose elements work in compression (Fig. 113*b*). Close to this case is the arched beam (Fig. 113*c*) working also mainly in compression.

The relationship between the flexure f_b of the simply supported beam in the plane of the acting bending forces and the flexure f_{tr} of the truss system (Fig. 113*b*) is given approximately by

$$\frac{f_b}{f_{tr}} = 1.3 \left(\frac{l}{d}\right)^2 \sin^2 \alpha \cos \alpha$$

where l = span of the beam;

d = diameter of the beam (and bars of the truss system);

α = side angle of the truss triangle

The f_b/f_{tr} ratio is plotted in the graph (Fig. 114) as a function of angle α for different values of l/d.

The graph indicates the large advantages of the system working in compression over the system subjected to flexure. In the acting load plane the beam flexure is hundreds and thousands of times greater than that noted for truss-type bar structures. Even with small values of angle α ($\approx 15°$), the flexure of the truss-type structure, for example, when $l/d = 50$, is 200 times less than that of the beam system subjected to flexure.

As in the preceding case (see Fig. 108) the trussed bar system has the highest rigidity when $\alpha = 45$-$60°$.

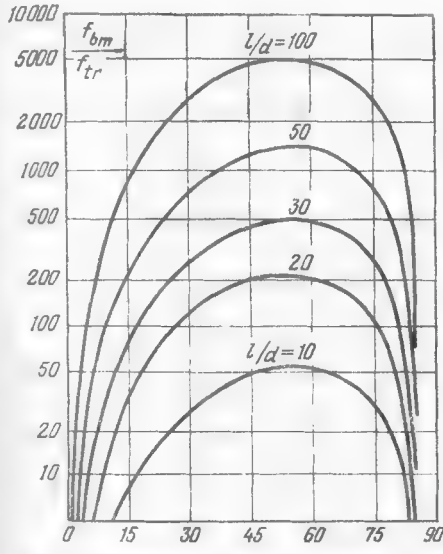


Fig. 114. Deflections of a beam and a truss system as a function of angle α

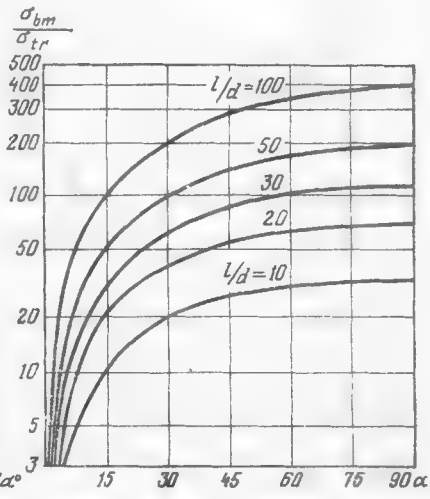


Fig. 115. Stress in a beam and rods of a truss system as a function of angle α

From the graph (Fig. 115), showing the relation between the maximum flexural stress σ_b in a beam and compressive stress σ_{tr} in a trussed bar system as a function of angle α for various l/d values, the stresses in the beam on average are 30-300 times larger than those in the truss system.

In rods having large l/d ratios working in compression the danger of buckling occurs. This feature must be considered during design.

Figure 116*a* shows loading of a cylinder. The force applied along the cylinder axis causes flexure of the end faces, transferred to the coarse through the connecting flange (the deformations are shown by dashed lines). The system is not rigid.

On changing the cylinder to a cone (Fig. 116*b*), by the main scheme of taken-up forces, it approximates the case of the truss bar system in Fig. 113*b*. The cone walls work mainly in compression, the role

of the sides withstanding the thrust in this case is fulfilled by the stiff circular sections of the body, thus limiting radial deformations of the walls. The highest rigidity with lowest weight is possessed by cones having vertex angles from 60 to 90°.

Improved rigidity is displayed by other components when their shapes are similar to a cone, for example, spherical (Fig. 116*c*), egg-shaped (Fig. 116*d*) constructions, etc.

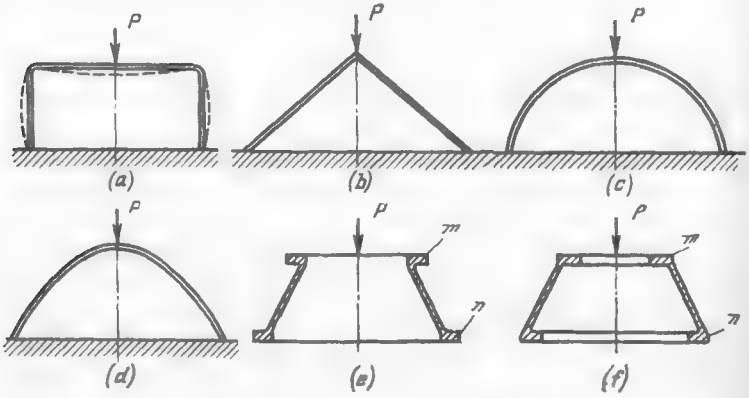


Fig. 116. Constructions operating in compression

As for the cone working in flexure, in order to improve the strength and rigidity of thin-walled components it is essential to use stiffening collars (Fig. 116*e*, *f*), of which the upper collar *m* takes up compressive loads and the lower one *n*—tensile loads.

(*b*) *Blocking the Deformations*

When expressed in most general terms, the problem of improving rigidity means finding the points of greatest deflection under the

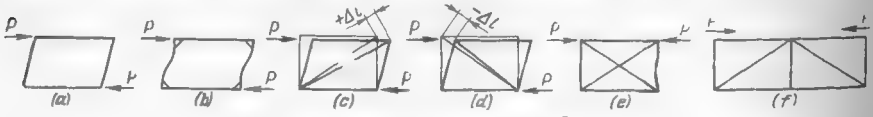


Fig. 117. Operation of diagonal struts

action of acting loads in the system and avoid these displacements by introducing tension-compression elements arranged in the line of the displacements.

A classic example of solving this problem is the rigidity improvement in frames and truss constructions by using diagonal struts.

Figure 117*a*, shows a bar frame subjected to shear forces *P*. The insignificant rigidity of this frame system is determined only by the flexural rigidity of vertical bars and the rigidity of the corner

joints. After some stiffening elements, for example, gussets, have been introduced into the system (Fig. 117*b*), the working system of the frame approximates that of the constrained beams and somewhat lessens the deformations.

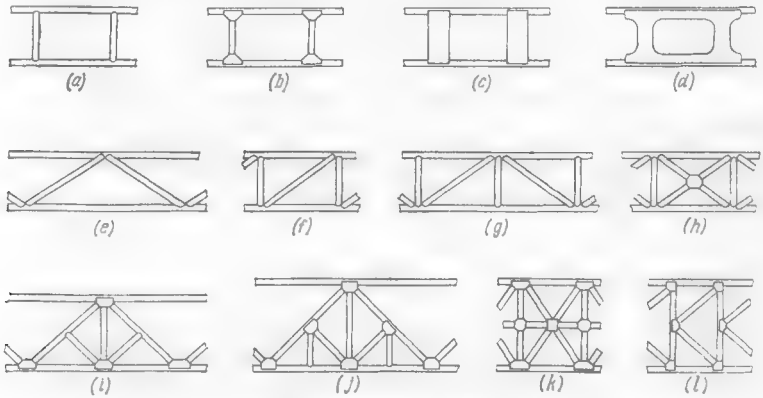


Fig. 118. Flat trusses

The most efficient way, however, is the introduction of diagonal struts, which work in tension or compression. A tensile strut (Fig. 117*c*), diagonally linking the frame elements, must extend by the value Δl as shown in the Figure. Owing to the inherent features of this form of loading, the small deformation values efficiently

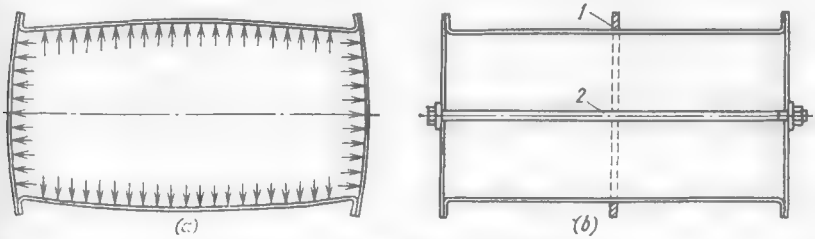


Fig. 119. Arresting deformations

stop frame distortion. A compression strut (Fig. 117*d*) will act similarly, however, in this particulas case it is necessary to consider possible buckling of the compressed bar, which in this case makes this system less desirable.

If the loads act alternately in both directions, Fig. 117*e*, the struts are either diagonally crossed or, Fig, 117*f*, in alternating directions.

Figure 118 shows various design schemes of simple trusses (girders) in order of rising rigidity (Fig. 118*a-h*), and also composite simple

trusses with stiffening elements preventing buckling and loss of bar stability (Fig. 118*i-l*).

The wall deformation of a cylindrical vessel under the action of internal pressure is shown in Fig. 119*a*. Parts *m* having the greatest deformations are advisably connected by tension elements (Fig. 119*b*): around the course by ring *1* and the end plates by anchoring bolt *2*.

(*c*) *Cantilevered and Double-Support Systems*

Should flexure for constructional and functional reasons be unavoidable, the decrease of deformations and flexural stresses must be the first task.

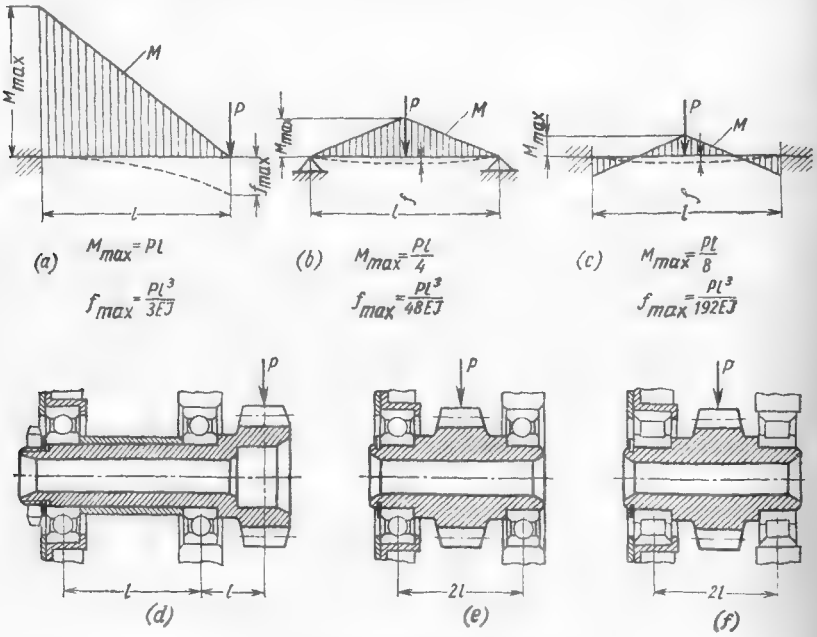


Fig. 120. Loading schemes

Beam flexures, most commonly encountered in practice, are given in Fig. 120: a cantilevered (overhanging) beam (Fig. 120*a*) a freely supported beam (Fig. 120*b*) and a constrained beam (Fig. 120*c*).

From the comparison of the maximum bending moment and flexures the advantages in rigidity and strength clearly lie with the freely-supported beam, than with the cantilevered one. With identical loads, beam lengths and sections, the maximum bending moment (and, hence, the maximum flexural stresses) with the simply-supported beam are four times, and with the constrained beam

eight times less than that of the cantilever. Even greater is the advantage in rigidity. The maximum flexure of the simply supported beam is sixteen times and that of the constrained beam sixty four times less than that of the cantilever.

In practice the difference between the simply-supported and cantilevered beams is not as sharp as illustrated. In comparable constructional variants of the cantilevered and simply-supported beam the length of a cantilever rarely equals the beam span distance, generally the cantilever length is very much less.

Let us consider a cantilevered (Fig. 120*d*) and simply-supported (Fig. 120*e*, *f*) arrangement of a pinion shaft. In this particular case, because of overall sizes, the span distance in the simply-supported arrangement cannot be less than $2l$, owing to which we have to compare these two alternatives on the basis of different and not equal lengths as in the preceding comparison.

The maximum bending moment for the simply-supported case is, in accordance with Fig. 120*e*, half that of the cantilevered arrangement. The gain in maximum flexural stresses is much greater since the resisting moment at the dangerous section (i.e., in the plane of the acting force P) of a simply-supported shaft is significantly greater than the resisting moment at the same section (i.e., in the plane of the leading bearing closest to the load) of the cantilevered shaft. For the relationships in Fig. 120, stresses in the critical section of the simply-supported shaft are 5 times less than those of the cantilevered shaft.

The maximum simply-supported shaft flexure is formally half that of the cantilevered shaft, but if the large moment of inertia I' of a simply-supported shaft, as compared to the moment of inertia I in a cantilevered shaft is considered, flexure will be still less (for the relationships in Fig. 120, the decrease will be 6.5 times).

The load on the bearings of the simply-supported shaft is less than the load on the front bearing of the cantilevered shaft $P\,(1 + l/L)$ by $2\,(1 + l/L)$ times.

The situation is better with the constrained shaft. The actual improvement in this case is obtained by increasing the rigidity of supports, e.g., employing roller bearings and strengthening the casing walls (Fig. 120*f*). Here the maximum bending moment is one quarter that of the cantilevered shaft, and half, when compared with the simply-supported shaft, mounted in ball bearings. The maximum flexure of the shaft, installed in rigid supports, is correspondingly less by 8 and 4 times (without considering the difference in I and I' values).

However, it is necessary to note that short and rigid shafts increasing support rigidity does not give practical advantage since the rigidity washes out the difference between the freely supported and constrained shafts.

(*d*) *Increasing Rigidity and Strength of Cantilevered Constructions*

If the use of a cantilevered component cannot be avoided, then its inherent disadvantages must be eradicated. First of all measures must be taken to shorten cantilever overhang and increase rigidity and strength of the construction's cantilevered portion.

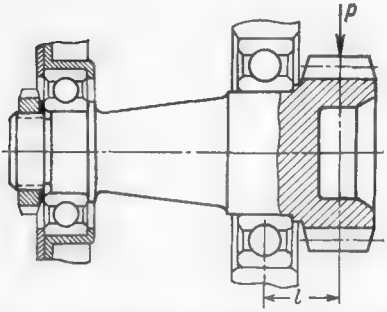


Fig. 121. Rational design of a cantilevered shaft

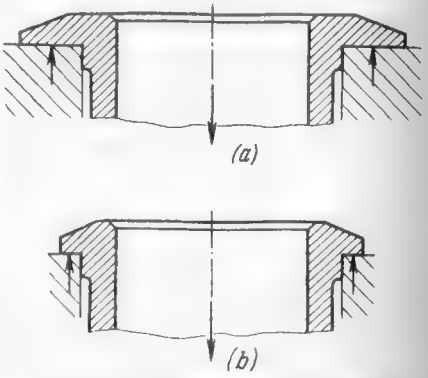


Fig. 122. Shapes of shoulders

The example in Fig. 121 shows an improved design of the cantilevered shaft presented in Fig. 120*d*. The cantilever length has been reduced to the limit permitted by the construction and the moments

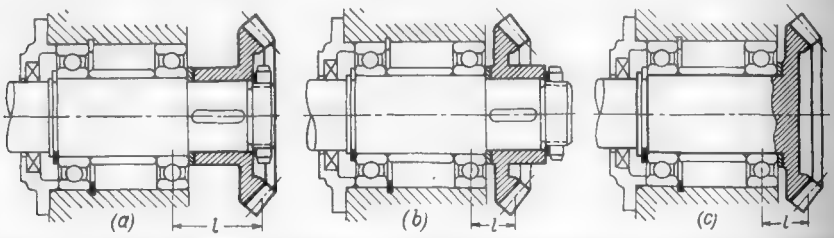


Fig. 123. Constructional methods of reducing overhang in cantilevered bevel gear assemblies

of inertia and resistance at the most loaded areas increased. The front bearing, accepting the highest load, is strengthened.

One of the commonest types of cantilevers, applied in mechanical engineering, is the shoulder of a supporting bush for a cylindrical component. In the incorrect design (Fig. 122*a*) the collar is excessively large. If the collar is reduced, for example, three times (Fig. 122*b*), the stresses across the critical section are decreased in the same proportion and the maximum deformations 27 times

In a number of cases it is possible to shorten the cantilever length by altering the shape of the part.

Example: The overhang of the bevel gear (Fig. 123*a*) can be decreased by altering the position of the gear hub relative to the rim (Fig. 123*b*) or by using a stem gear (Fig. 123*c*).

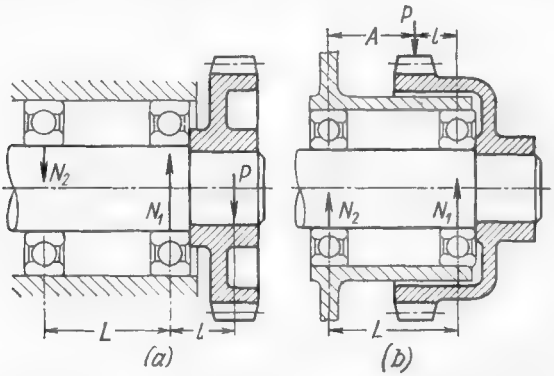


Fig. 124. Determining forces acting upon supports
(*a*) cantilevered shaft; (*b*) inverted cantilever

The loads on the cantilevered shaft bearings depend upon the distance L between the supports and cantilever length l (Fig. 124*a*). Thus, the load on the front bearing

$$N_1 = P\left(1 + \frac{l}{L}\right)$$

and that on the rear bearing

$$N_2 = P\frac{l}{L}$$

where P is the force acting on the cantilever.

From the graph, showing the dimensionless ratio N_1/P and N_2/P as a function of L/l (Fig. 125*a*), it is seen that the bearing loads sharply rise as the support span distance decreases. With the increase in the $\frac{L}{l}$ ratio the bearing loads fall; then N_1 asymptotically tends to the P value and N_2, to zero. When $L/l > 2$-2.5 the bearing load becomes practically constant, but when $L/l < 1$ it sharply rises. Thus, the best recommended range for the L/l ratios is 1.5-2.5 (shown hatched in Fig. 125*a*).

For a general rule it may be assumed that the distance between the supports must be twice the cantilever length. Naturally, higher L/l ratios have the advantage of fixing the shaft more accurately.

Figure 125*a*, shows as a function of L/l the load ratios of the front and rear bearings $N_1/H_2 = 1 + L/l$, which can be used as a guide when selecting bearings for those cases when their life expectancy has to be the same. Thus, the recommended values are $L/l = 2$, $N_1/N_2 = 3$.

Permissible loads on antifriction bearings are determined from the formula

$$Q = \frac{C}{(nh)^{0.3}}$$

where C = bearing work capacity coefficient;
n = speed, rpm;
h = bearing service life, h

Since n = const, and h = const, the work capacity coefficients of the front and rear bearings, when $L/l = 2$, will be within the ratio $C_1/C_2 = 3$.

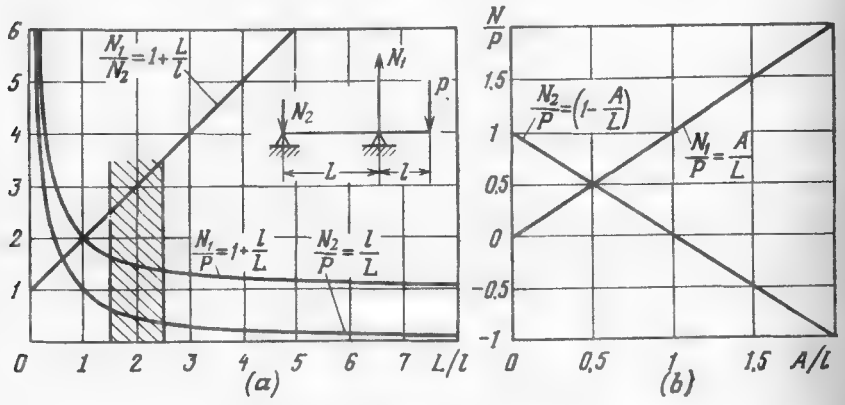


Fig. 125. Loads on supports
(*a*) cantilevered shaft; (*b*) inversed cantilever

Often an inverted cantilever is employed. In this case (Fig. 124*b*), the gear, so that load is applied in the span between the supports, is given a bell-shaped form; it thus works as an inverted cantilever.

The values of the bearing loads are shown in the graph, Fig. 125*b*, as dimensionless ratios N_1/P and N_2/P against A/L (A—distance from the rear bearing to the plane of force P). The position of the inverted cantilever must be within $A/L = 0$ to 1; if $A/L > 1$ the system becomes a direct cantilever.

It is evident from the graph that the maximum values of N_1 and N_2, in the region of the inverted cantilever, are equal to the acting force P (N_1/P and N_2/P ratios are equal to unity). Loads N_1 and N_2 have identical and equal values $= 0.5P$ when $A/L = 0.5$, when the force plane lies midway between the supports.

A second constructional example of an inverted cantilever is illustrated in Fig. 126*a*, *b*.

To fully eliminate the cantilever, the part is mounted on a stationary support *1* (Fig. 126*c*). Through this passes the flexure-relieved driving shaft, which imparts to the part through a splined rim

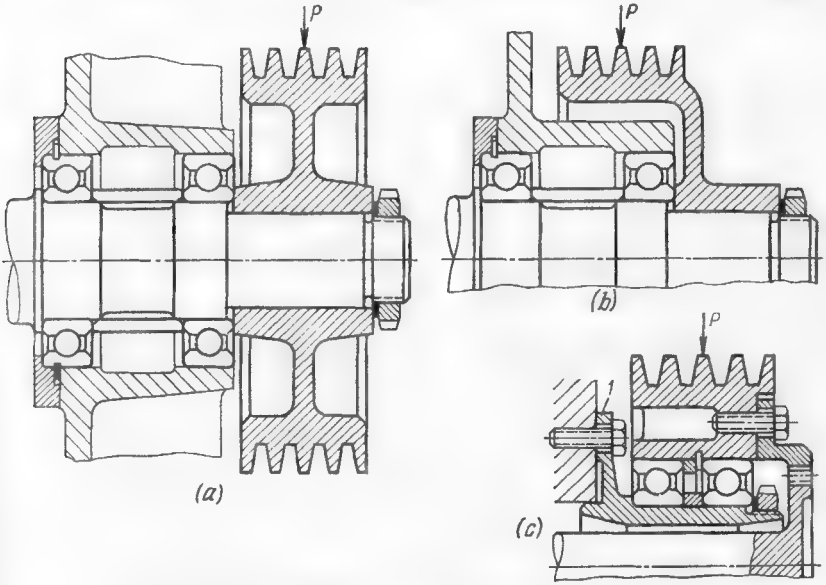


Fig. 126. Decreasing loads on cantilever supports

a pure torque moment. The bearings are loaded as in the simply-supported shaft. However, their working conditions are less favourable since the outer race rotates (and not the inner one as in the simply-supported shaft), due to which their durability period is shortened.

The above-described disadvantages of cantilevered systems by no means deprive the designer of their use. Cantilever systems are fully lawful elements of the construction and they are widely applied in practice. It is only necessary to know their peculiarities and eliminate the disadvantages by suitable design measures.

The use of cantilevers often assures more simple, compact, technological convenient construction for assembly than the double-supported shaft assemblies. Shown as an example in Fig. 127 are the constructions of two centrifugal pumps: the first (Fig. 127*a*) has the simply-supported shaft and the second (Fig. 127*b*) the cantilevered mounting.

The cantilevered version simplifies assembly, provides easier access to the vane and hydraulic cavity, betters the entry of operat-

ing liquids to the impeller, eliminates one seal and improves shaft alignment. Furthermore, shaft supports are accommodated inside one housing and the bearing fitting holes can be machined at one setting.

In the simply-supported arrangement the supports are aligned to each other through the housing joint, the two halves of which

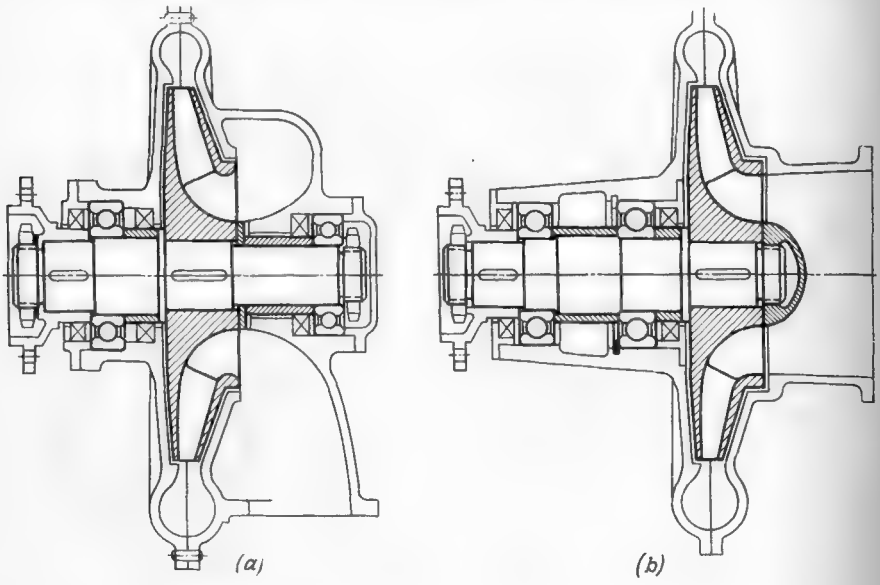


Fig. 127. Centrifugal pump with (*a*) simply supported and (*b*) cantilevered shaft

are, owing to the unit's design, fixed only by fitted dowels; in other words, shaft fitting holes cannot be machined at a time.

Taken as a whole, the cantilevered design strongly gains in simplicity, accuracy of manufacture, reliability and operation convenience.

(*e*) *Rational Arrangement of Supports*

The flexure of a simply-supported beam is proportional to the third power of the span, therefore reducing span distance—an effective means of enhancing rigidity.

Figure 128 shows a simply-supported gear assembly. If the span between the supports is reduced three times, then the maximum bending moment and stresses in the shaft are also reduced three times and the maximum flexure by 27 times. When the shaft diameter $d = 40$ mm, length $L = 200$ mm and load $P = 1000$ kgf, the shaft flexure, Fig. 128*a*, will attain a comparatively large value (of the

order of 0.1 mm), disadvantageous for gear operation. After decreasing the span three times (Fig. 128*b*), the flexure reduces to a negligible value (of the order of 0.004 mm).

Very often the rigidity of a system is improved by the introduction of additional supports (Fig. 129).

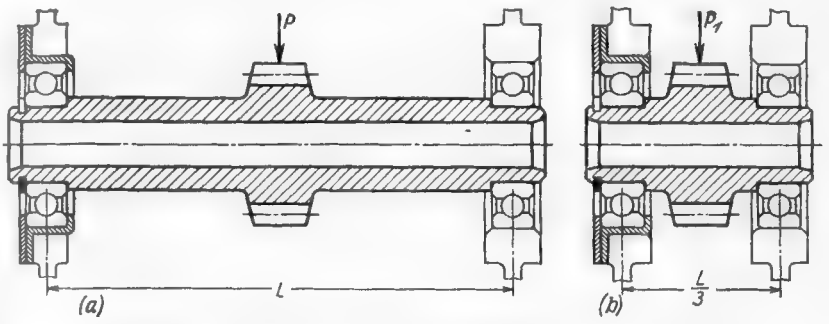


Fig. 128. Reducing span distance between supports

In the construction, shown in Fig. 129*a*, the crankshaft is mounted in two bearings. The system has low rigidity; to increase its value it is necessary to enlarge the web and neck sections of the shaft.

The rigidity characteristic can also be improved by means of an additional central support (Fig. 129*b*) or, the more so, by several supports (Fig. 129*c*). The last version is now almost always used.

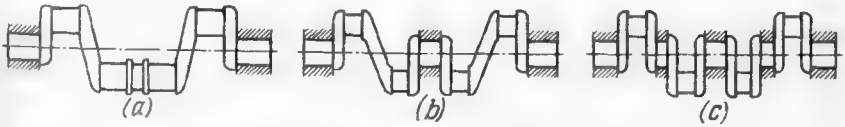


Fig. 129. Arrangement of crankshaft supports

Figure 130 shows schematically methods of increasing rigidity and strength of the fastening assembly of a connecting rod and fork. Since the connecting rod generally oscillates about the fork end at small swings it is possible to introduce additional supports which practically eliminate flexure.

The original, rather popular design, presented in Fig. 130*a*, possesses poor rigidity, since the pin here is subjected to flexure.

In the design, illustrated in Fig. 130*b*, the pin is relieved of flexure owing to a thrust-pad, provided in the fork.

Flexure can be sharply decreased by increasing the length of the connecting rod upper bearing surface (Fig. 130*c*, *d*). Over section *h* the pin works in compression. Since compressive deformations are

infinitesimal in comparison to those in flexure, practically all the load taken up by the pin is a compressive one.

The designs, depicted in Fig. 130*b-d*, are suited for mostly unilateral loads, acting in the direction shown by an arrow. Moreover, in these designs the amplitude of the connecting rod's oscillations about the fork is limited.

In the designs, intended for loads in both directions with large oscillation amplitude, strengthening is obtained by increasing the number of supports and lessening span distance over which the

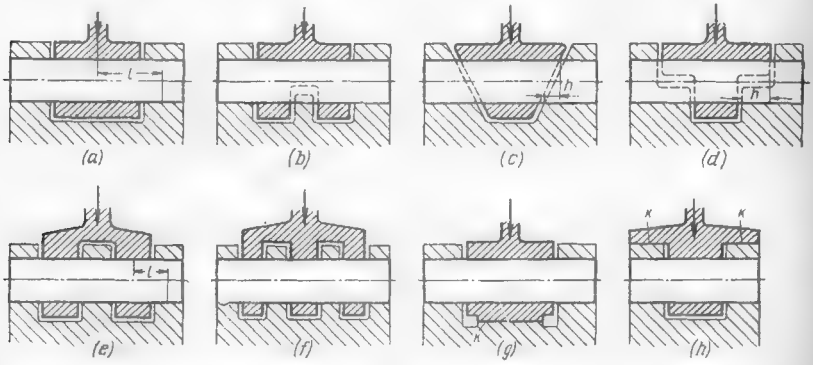


Fig. 130. Increasing rigidity of a connecting rod and fork assembly

flexural loads are taken (Fig. 130*e*). In this design in view of the doubled shortening shoulders l the force of the flexural stresses are also decreased by half, and the deformations, by eight times as compared with the original design (Fig. 130*a*).

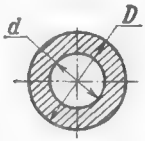
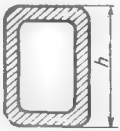
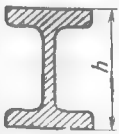
With the increased number of supports (Fig. 130*f*), the loading scheme approximates pure shear. Changing to shear and increasing the number of thus loaded sections significantly adds to the strength and rigidity of the assembly.

In some instances, when dealing with unilateral loads, acting forces can be transmitted directly to the supports, fully relieving the pin of load (Fig. 130*g*, *h*). The manufacture of such constructions is more complex than the previous ones, since here it is necessary to precision-machine the cylindrical bearing surfaces k coaxially with the pin bearing surfaces. Otherwise, the scheme of applied forces becomes indeterminate.

(*f*) *Rational Sections*

It is important that the increase in rigidity is not accomplished by increasing weight of the detail. In the general form, the solution to the problem means strengthening sections, which under the given loads are subjected to the highest stresses, and removing weight

Comparative Rigidity/Strength Characteristics of Differently Shaped Sections Having the Same Weight

Section	Ratios		I/I_0	W/W_0
	$\frac{d}{D}$	—	1	1
		0.6	2.1	1.7
		0.8	4.5	2.7
		0.9	10	4.1
	$\frac{h}{h_0}$	—	1	1
		1.5	3.5	2.2
		2.5	9	3.7
		3.0	18	5.5
	$\frac{h}{h_0}$	—	1	1
		1.5	4.3	2.7
		2.5	11.5	4.5
		3.0	21.5	7.0

from the unloaded or slightly loaded areas. In flexure the stressed sections are those farthest from the neutral axis. In torsion the external fibres are mostly stressed; moving radially and inwards, the stresses become weaker and at the centre are zero. Consequently, for these cases it is more rational to concentrate material at the peripheries and remove it from the centre.

Generally the greatest rigidity and strength characteristics with smallest weight are possessed by components with thin-walled sections, i.e., parts such as box sections, tubes and shells.

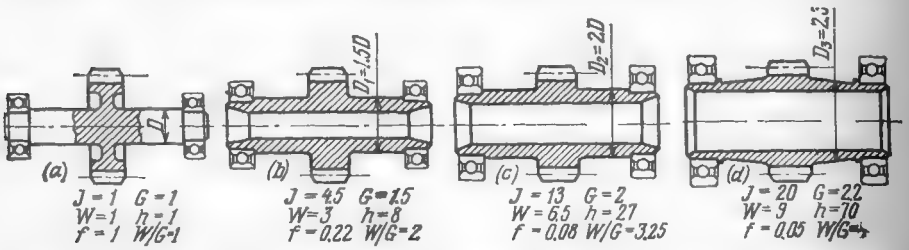


Fig. 131. Effect of increase of shaft diameter upon the rigidity, strength and weight of construction and durability of antifriction bearings

Table 17 gives rigidity and strength comparisons for differently-shaped sections. The base of the comparison depends upon similar weight conditions of parts, expressed as similar cross-sectional areas. The strength and rigidity improvements are obtained by successful application of the material distribution principle in the regions of the highest acting stresses.

For cylindrical sections the moment of inertia I_0 and the moment of resistance W_0 of a solid round section are taken as the units of comparison; with respect to the other parts, a solid square-shaped section.

The dependence between weight, strength and rigidity of cylindrical shafts with different d/D ratios is described in a general form in Figs. 36, 37 and 39.

A constructive example of a gear and integral shaft running in antifriction bearings is given in Fig. 131. The parameters of rigidity I, strength W, weight G and durability h of bearings are given for successive increases of shaft diameters (and sizes of bearing supports).

Unit characteristics are those of the solid shaft (Fig. 131*a*).

(*g*) *Improving Transverse Rigidity*

Together with the increase in external sizes and thinner wall sections it is necessary to increase rigidity in the transverse direction of acting bending forces in order to avoid constructional stability losses.

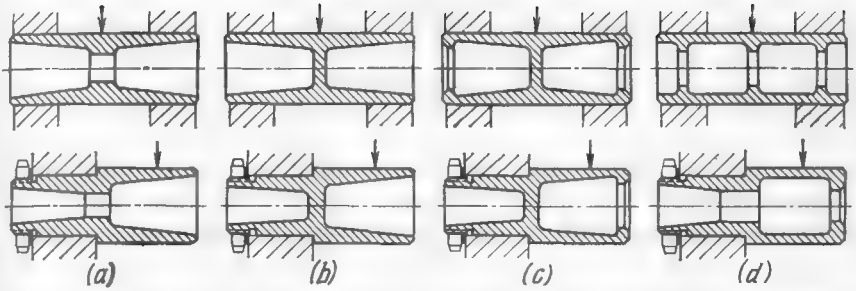


Fig. 132. Methods of increasing the radial rigidity of hollow components

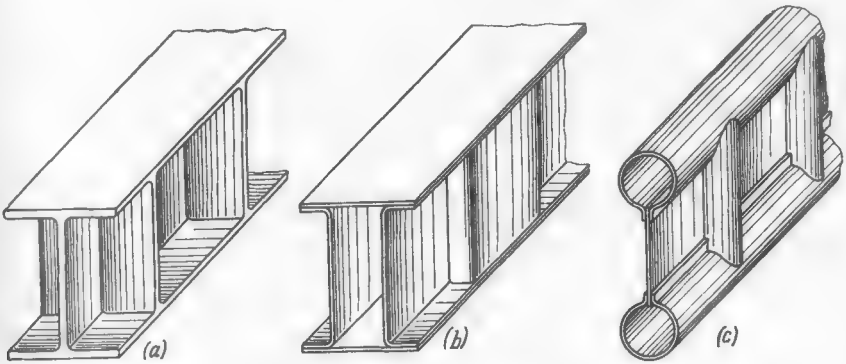


Fig. 133. Increasing the rigidity of beams

(*a*) by stiffening partitions; (*b*) by stiffening boxes; (*c*) by semi-circular stiffening elements

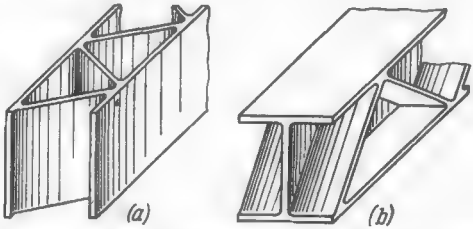


Fig. 134. Beams reinforced with diagonal stringers

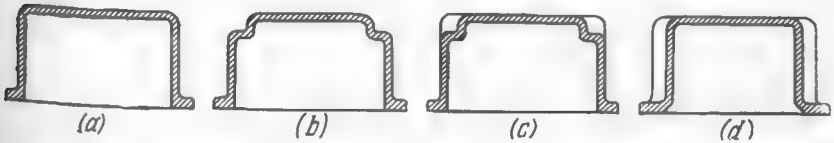


Fig. 135. Increasing the rigidity of shapes by local section stiffening

For cylindrical shafts the problem is solved by using stiffening collars and webs (Fig. 132*a*, *b*). Stiffening collars are advisably positioned in the plane of acting loads, on bearing, fixing areas and also at the free ends of a part (Fig. 132*c*, *d*).

Figure 133 shows reinforcement of beams by transversal ribs and stiffening boxes.

Snake-like diagonal stringers in the form of webs strongly improve rigidity (Fig. 134*a*, *b*), and also local section stiffening (Fig. 135). Thus, constructions with longitudinally formed stiffening rib angles at the transition points where vertical walls change to horizontal ones (Fig. 135*b*) have greater rigidity than the original construction (Fig. 135*a*) in spite of the formal lessening of inertia moment. The rigidity parameter increases also when the longitudinal rib has transverse stiffening ribs spaced over the part length (Fig. 135*c*, *d*).

Table 18 illustrates how longitudinally arranged webs affect the rigidity of profiles during flexure and torsion. Diagonal webs have the strongest effect. One diagonal stringer will suffice; another stringer will enhance the rigidity but to a small degree.

Table 18

Increasing Rigidity of Sections by Longitudinal Webs

Profile	Factors				
	I_{flex}	I_{tors}	G	$\frac{I_{flex}}{G}$	$\frac{I_{tors}}{G}$
	1	1	1	1	1
	1.17	2.16	1.38	0.85	1.56
	1.55	3	1.26	1.23	2.4
	1.78	3.7	1.5	1.2	2.45

(h) Ribbing

Ribbing finds wide application in improving rigidity, particularly of cast housing-type components. Due care, however, should be exercised when employing this technique since wrongly related ribbed sections and ribbed details may weaken the part instead of strengthening it.

If a part has external ribs and is subjected to flexure acting in the plane of ribs (Fig. 136*a*), then considerable tensile stresses may occur

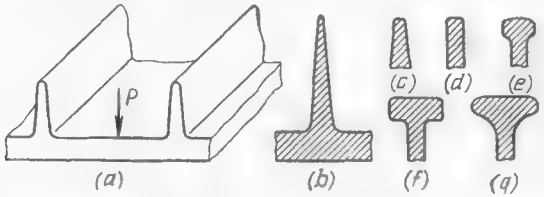


Fig. 136. Rib forms (in the order of increasing strength)

at the rib ridge, attributable to the comparatively small rib width and cross sectional area. The situation is aggravated when thin ribs, tapering to their tip, are used (Fig. 136*b*, *c*); the failure of parts always begins at the rib tip. Strength is significantly increased when thickening the rib, particularly at the critical region, i.e., the rib tip (Fig. 136*d-g*).

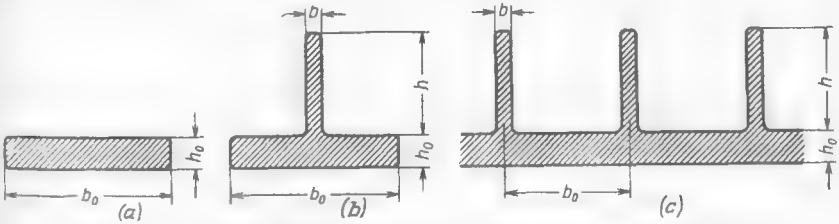


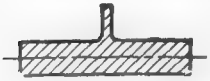
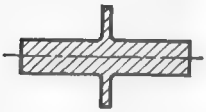
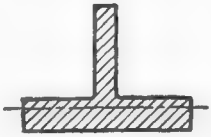
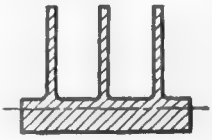
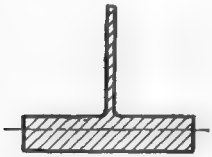
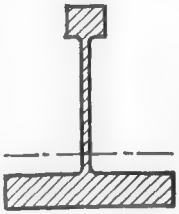
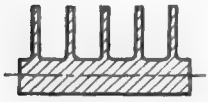
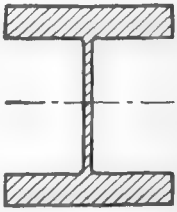
Fig. 137. Effect of ribs on the rigidity and strength of shapes

Weakening of details by ribs is formally expressed by the reduction of the details moment of resistance. Table 19 shows how stiffening ribs affect the moments of resistance and inertia of a rectangular section. The moment of resistance of a non-ribbed rectangular section is taken as unity.

The influence of relative height and width of ribs upon the rigidity and strength of a part is easily expressed in a generalized form. Compare the strength and rigidity of a rectangular section profile (Fig. 137*a*) with the similar profile provided with a rib (Fig. 137*b*).

Calculations show that correlation between the moment of inertia I of a ribbed profile and the moment I_0 of the original profile

Effect of Ribs Upon Strength and Rigidity of Parts

Section	Factors		Section	Factors	
	W	*I*		*W*	*I*
	1	1		2	21
	0.5	1.06			
	0.6	1.2		2	11
	0.6	2		2	11
		7.8		5	35
	1	4.5			
	1	4.5		180	300

is expressed by the relation

$$I/I_0 = 1 + \delta\eta^3 + 3\delta\eta\,(1 + \delta\eta)\left(\frac{1+\eta}{1+\delta\eta}\right)^2 \qquad (4.29)$$

where $\eta = h/h_0$ = ratio of rib height h to original profile height h_0
$\delta = b/b_0$ = ratio of rib width b to original profile width b_0

When related to sections with a series of parallel ribs (Fig. 138*c*), the reciprocal of δ will be the relative rib pitch i_0, i.e., the ratio of rib pitch to rib thickness

$$t_0 = b_0/b$$

In this case the value δ can be termed as the density of rib spacing.

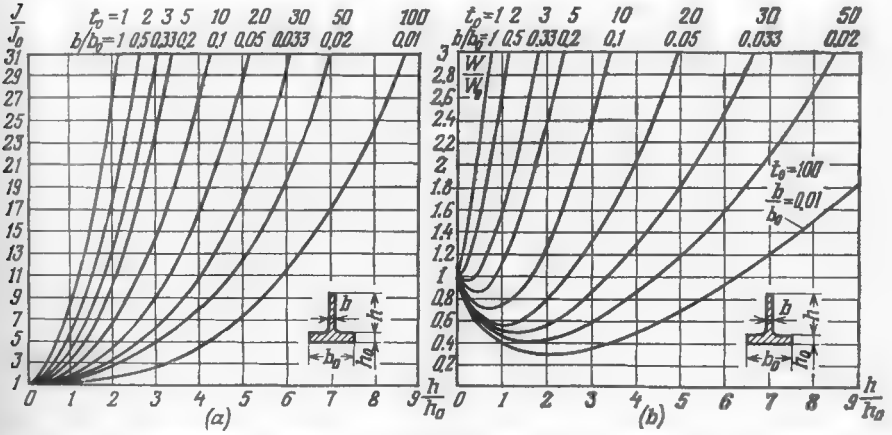


Fig. 138. Rigidity and strength of ribbed sections as a function of relative rib height h/h_0 for different values of relative rib width b/b_0

(*a*) rigidity; (*b*) strength

The correlation of moments of the compared profiles is equal to

$$\frac{W}{W_0} = \frac{I}{I_0} = \frac{1+\delta\eta}{1+2\eta+\delta\eta^2} \qquad (4.30)$$

On the basis of formulae (4.29 and 4.30) are constructed the graphs (Fig. 138), which show the relative influence of rib dimensions on strength and rigidity.

As evident from the graph (Fig. 138*a*), the introduction of ribs in all cases enhances the moment of inertia of a section and, consequently, the flexural rigidity of a part. The higher the rib and the greater its relative thickness the better the rigidity.

The picture is quite different when dealing with the moments of resistance (Fig. 138*b*). Introduction of ribs, whose section is small in comparison with that of the part to be ribbed (small values of

h/h_0 and b/b_0, large pitch), worsen the moment of resistance, i.e. weakens the part. Under unfavourable conditions ($h/h_0 = 2$; $b/b_0 = 0.01$) the moment of resistance will drop to a third of that of the original profile.

The picture becomes more expressive if we plot the relative pitch values t_0 on the abscissa and variations of the moment of resistance, for different values of relative rib height h/h_0, on the ordinate (Fig. 139). The portions of curves, located below the line $W/W_0 = 1$,

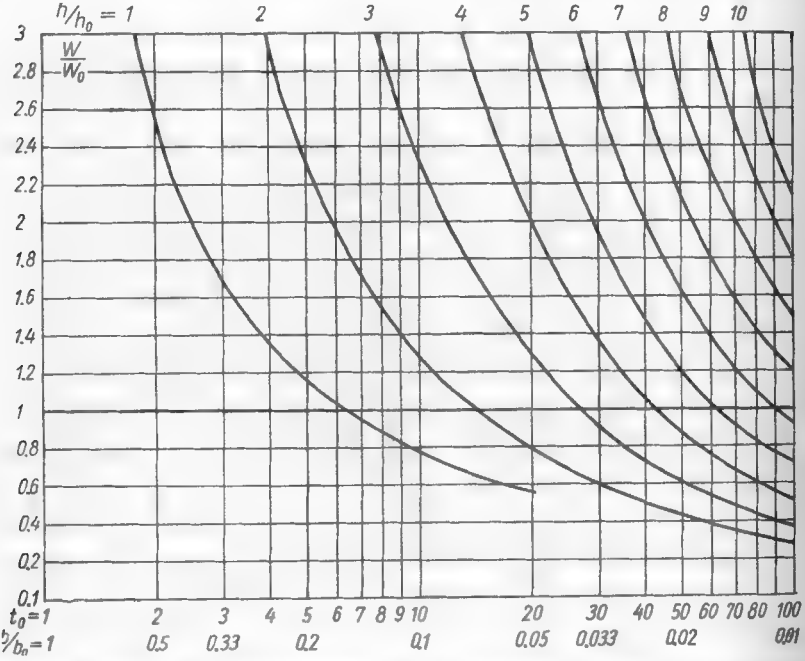


Fig. 139. Strength of ribbed sections as a function of relative rib pitch $t_0 = b/b_0$ for different values of relative rib height h/h_0

describe relationships at which strength begins to fall (large relative pitches t_0, small relative rib thicknesses b/b_0). The less the rib height, the sharper the weakening is expressed.

Weaking may be prevented by increasing rib height. Ribs, whose relative height $h/h_0 > 7$, will not worsen strength of parts up to the largest values of relative pitch encountered in practice ($t_0 = 100$).

However, for cast components increases in rib height are limited by the casting technique. In practice the relative height rarely exceeds $h/h_0 = 5$. Casting technique also restrict rib thicknesses. Generally thickness is kept within (0.6-0.8) h_0.

Actually the other way is to decrease relative pitch. When $t_0 > 6$ weakening does not begin even with the smallest ribs ($h/h_0 = 1$).

From the graph, plotted in Fig. 139, it is possible to find the pitch values of different height ribs which do not weaken the part. These values correspond to abscissas of points, in which curves h/h_0 intersect the ordinate $W/W_0 = 1$.

If relative pitches t_0 are expressed as a function of relative rib height h/h_0, then the curve $W/W_0 = 1$ (Fig. 140) will correspond to

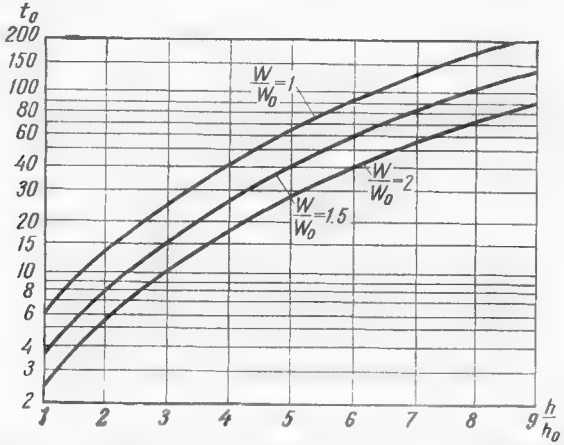


Fig. 140. Relative rib pitch t_0 as a function of relative rib height h/h_0 for different ratios W/W_0

the case when strength of a part is not affected by the increased number of ribs and the curves $W/W_0 = 1.5$ and $W/W_0 = 2$ will correspond to the cases, when strength of a part rises in proportion to the increased number of ribs.

For practical determination the maximum allowable pitch is found from the relation

$$t_0 = \frac{b_0}{b} < 2\left(\frac{h}{h_0}\right)^2 \tag{4.31}$$

expressed as the averaged t_0 values when

$$W/W_0 = 1.5\text{-}2$$

The maximum admissible rib pitch

$$t = 2b\left(\frac{h}{h_0}\right)^2 \tag{4.32}$$

where b is the rib thickness, mm.

Based on Eq. (4.32) a graph (Fig. 141) is constructed which allows limiting t values to be found for different rib parameters.

It should be noted that if stresses in a part are insignificant, usually the case in housing-type components, then the decrease in strength due to the introduction of unfavourably shaped ribs is not dangerous.

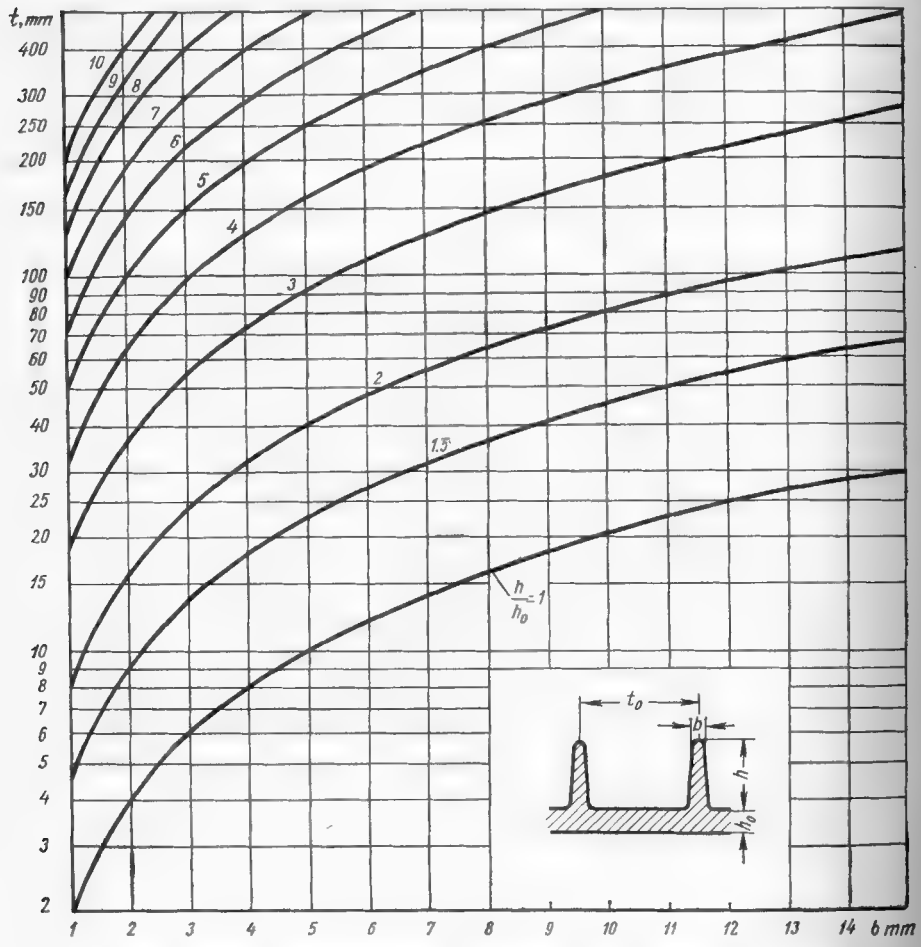


Fig. 141. Graph for determining the maximum admissible rib pitch

In such cases the designer may freely apply ribs (including those, working in tension), as a means of enhancing rigidity, without considering the lessening strength.

For a heavily loaded part all the above recommendations remain valid and are important for correct design practice.

Triangular ribs. Very often use is made of ribs, whose height diminishes in the plane of the bending moment (triangular-shaped ribs).

With such rib forms whatever their initial height sections are inevitable where weakening of the detail begins.

Figure 142 pictures typical shapes of triangular ribs on a cylindrical cantilevered part being bent by a force applied at the cantilever end. Under each figure is shown a qualitative picture of changes in the moment of resistance W and flexural stresses σ along the part

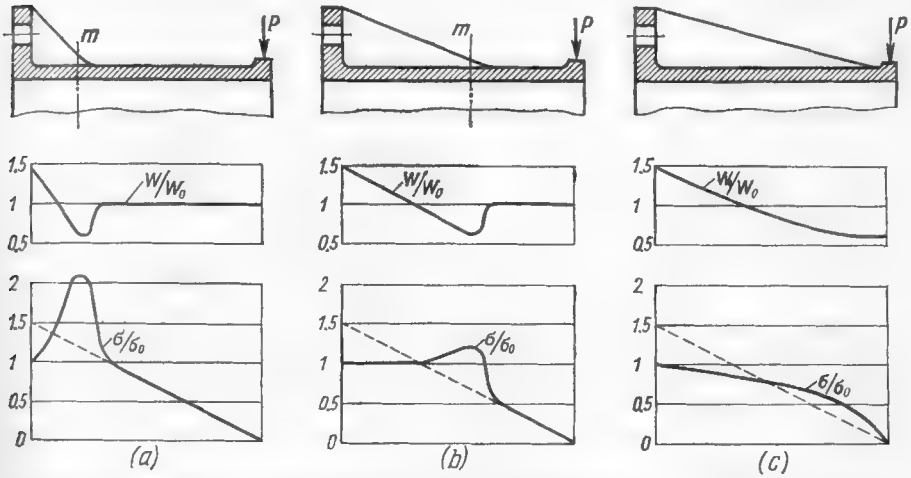


Fig. 142. Influence of triangular ribs on the strength of a cantilevered part

axis. For the unit moment of resistance the moment of resistance W_0 of a non-ribbed portion of a part is taken; for stresses, the value of the basic cantilever stress σ_0, i.e., at the point where the cylinder joins the flange. Stress values for the non-ribbed part are indicated by dashed lines.

For the rib form, shown in Fig. 142*a*, weakening begins at the part *m* where the rib joins the cylindrical wall. Such a rib form is particularly unfavourable as weakening occurs at the region of large bending moment values and at the weakened part a sharp rise in stresses takes place.

Somewhat better are long ribs (Fig. 142*b*). The weakened section *m* is displaced to the region of smaller bending moment. Stresses at the weakened section slightly exceed maximum stresses in the part.

The most advantageous case is when the rib extends to the cantilever end (Fig. 142*c*). Here the weakened area is at the region of minimum bending moment and almost has no effect upon stress values.

Ribs in compression. Internal ribbing. The relationships for ribs undergoing tension during flexure (Fig. 137) formally hold for ribs working in compression during bending (Fig. 143). Ribs, subjected

to compression, work in more favourable conditions since the majority of cast materials resist compression rather better than tension. With the introduction of ribs the phenomenon of the fall in strength working in compression has less significance than when working in tension.

The better strength inherent in ribs working in compression as well as the wide free choice of their parameters compels, in all cases, the preference to return to ribs in compression than those in tension.

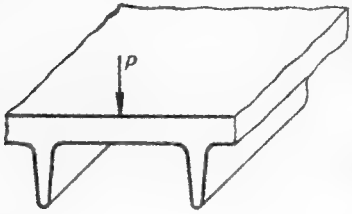


Fig. 143. Flexure of a cantilevered member with internal ribs working in compression

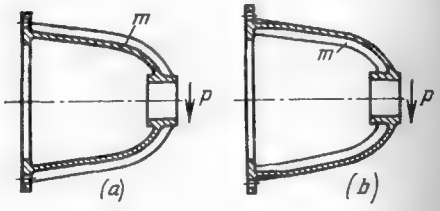


Fig. 144. Housing component
(*a*) external ribs; (*b*) internal ribs

This means that in housing-type components preference is given to internal ribbing and not external ribbing.

For a housing, having external ribbing loaded with bending force P (Fig. 144*a*) the main part of the load is taken by ribs m, positioned away from the side of acting load, and also by the housing walls (especially those portions parallel to the bending moment plane). The opposite ribs, which in accordance with the force scheme should work in compression, hardly take part, because the load reaching them is strongly reduced. Hence, in this loading mode the ribs work largely in tension, i.e., unfavourably.

With internal ribbing (Fig. 144*b*) the ribs m positioned on the side of the acting load work in compression. The opposite ribs, which according to the loading scheme should work in tension, are practically free of load.

Apart from the improved rib strength, the internal ribbing allows the strength and rigidity of the housing as a whole to be sharply increased, as the radial dimensions of the housing walls can be increased. Within the same overall sizes (being determined by the rib outline for an externally ribbed housing) one can obtain substantial improvement in the values of the moment of resistance and the moment of inertia.

The moulding of internal ribs is simpler (particularly when the part interior is formed by cores). Furthermore, internal ribbing

betters the outside appearance of the components. In general, internal ribbing is preferred in all cases, except in special ones (when outer fins, for example, are necessary for cooling purposes).

Design Rules

When designing ribs it is necessary to observe the following basic rules:

avoid loading ribs in tension, apply in all cases, when permitted by the design, ribs working in compression;

for strength reasons be sure that the height of ribs, intended for housing-type component, is not below (8-10) h_0 (h_0—wall thickness), if the ratio between ribs total thickness and wall width is small (i.e., of the order of $b/b_0 = 0.01$). Should such a height be unobtainable, owing to size or casting conditions, then rib pitch must be increased according to Eq. (4.32);

ribs of triangular form must extend to the plane of acting bending forces;

ribs must extend to rigidity points, around fastening bolts in particular (Fig. 145).

It is advisable to thicken rib ridges, because ribs with sharp tips invite higher stresses on their edges.

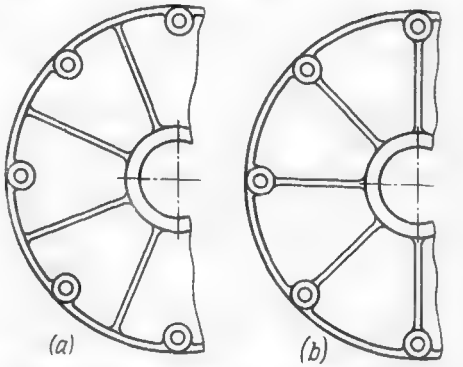


Fig. 145. Arrangement of ribs on a part used as a cover
(*a*) poor design; (*b*) good design

Occasionally it is most advantageous to obviate ribs at all: this will even add to the part strength.

Ribbing of parts subjected to torsion. When loading cylindrical parts, or other similar shapes, by a torque moment longitudinal straight ribs insignificantly increase rigidity (Fig. 146*a*). Such ribs quickly become harmful since they are subjected to flexure (in the plane normal to the rib edge), which causes higher stresses in the ribs.

During torsion it is more favourable to use diagonal ribbing (Fig. 146*b*), which under the influence of the torsional moment work in compression and substantially increase rigidity. The described design is a particular case of applying the principle of diagonal stringers.

The design, depicted in Fig. 146*b*, is calculated for a torque of one constant direction. For an alternating torque moment or a snake-like pattern (Fig. 146*d*) crossed ribs (Fig. 146*c*) are advisable.

Diagonal and helical ribs are less prone to internal stresses, which occur during casting shrinkage due to its non-uniform setting.

However, the moulding of diagonal ribs on outer cylindrical, conical and the like surfaces is difficult.

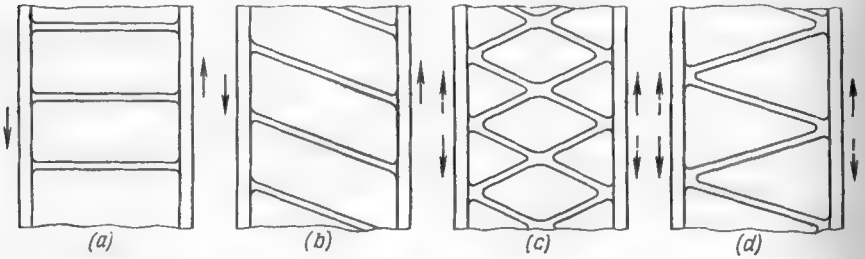


Fig. 146. Rib forms on a cylindrical component working in torsion

Therefore, when dealing with cylindrical and similar-shaped components, working in torsion, it is advisable, as in the case of flexure, to use internal ribbing.

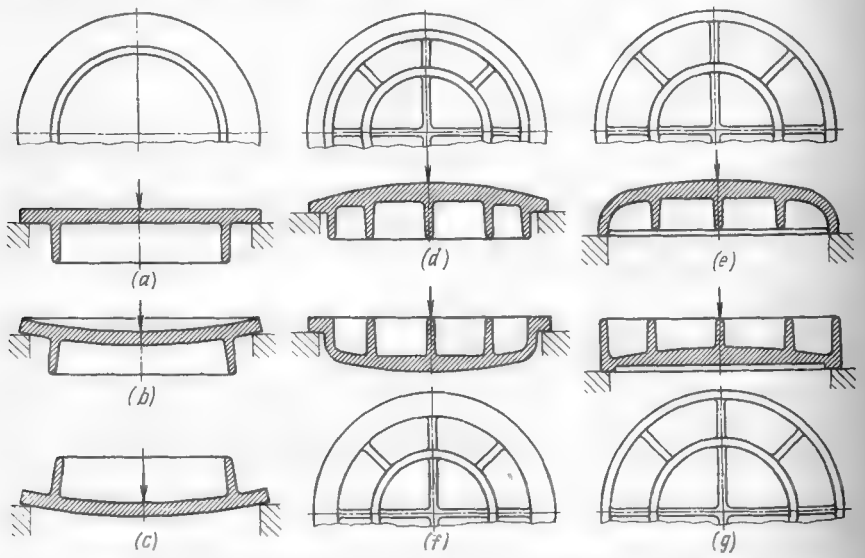


Fig. 147. Circular ribs on a disk subjected to flexure

Circular ribs. Circular ribs, along with the usual straight ribs, are used to enhance the rigidity of round components, such as disks, cylinder ends, etc.

The mode of action of these ribs is peculiar. Let us assume for example, that a round plate with a circular rib is bent by some axial force, applied in the centre (Fig. 147*a*). The deformations of the

plate are imparted to the circular rib; its walls tend to extend towards the periphery (Fig. 147*b*). The tensile stresses, arising in the ring, check the plate flexure.

The circular rib, facing the load (Fig. 147*c*) will act in a similar way, the difference being that it is subjected to inward radial compression.

To impove rigidity it is recommended that the circular ribs be made higher and positioned at the radius, where the sag angle of

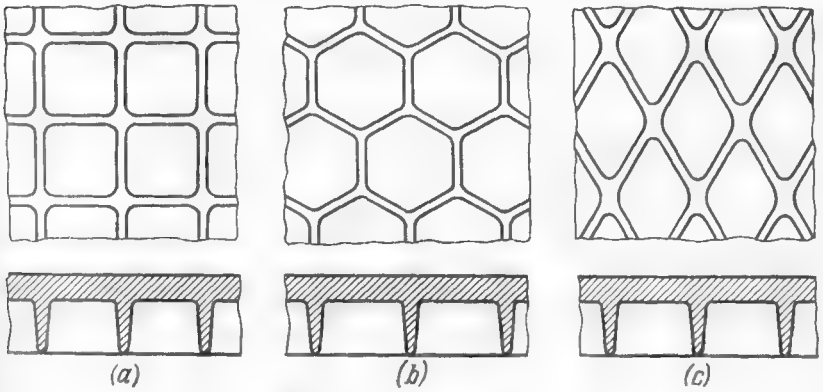


Fig. 148. Rib forms

the plate has its greatest value: for plates, resting on their rim, it will be closer to the periphery and for plates with built-in rims, closer to their mean radius. Positioning ribs at relatively small distances from the plate centre is almost useless.

The most effective combination of circular and radial ribs are shown in Fig. 147*d-h*.

Structural varieties of ribs. To impart especially high rigidity the following types of ribs are employed: wafer (Fig. 148*a*), honeycomb (Fig. 148*b*) and diamond (Fig. 148*c*).

Often use is made of hollow ribs (Fig. 149), open relief (*1-9*, *13*) or closed profile (*10-12-14*). In contrast to conventional ribs hollow ribs under all circumstances enhance the rigidity and strength of constructions.

Closed ribs are more rigid than the open ones, but their moulding is more difficult. Practically the same results are obtained when applying open ribs, strengthened by cross partitions (*3*, *6*, *9* and *13*).

Internal hollow ribs (*13*, *14*) are more preferable than external ones. In the range, where rectangular-shaped internal closed ribs join each other, the most rigid and strongest double-walled box-section construction is obtained (*15*).

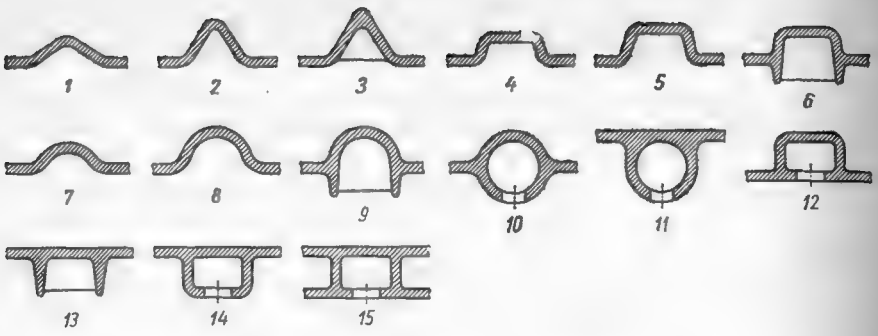


Fig. 149. Hollow rib profiles

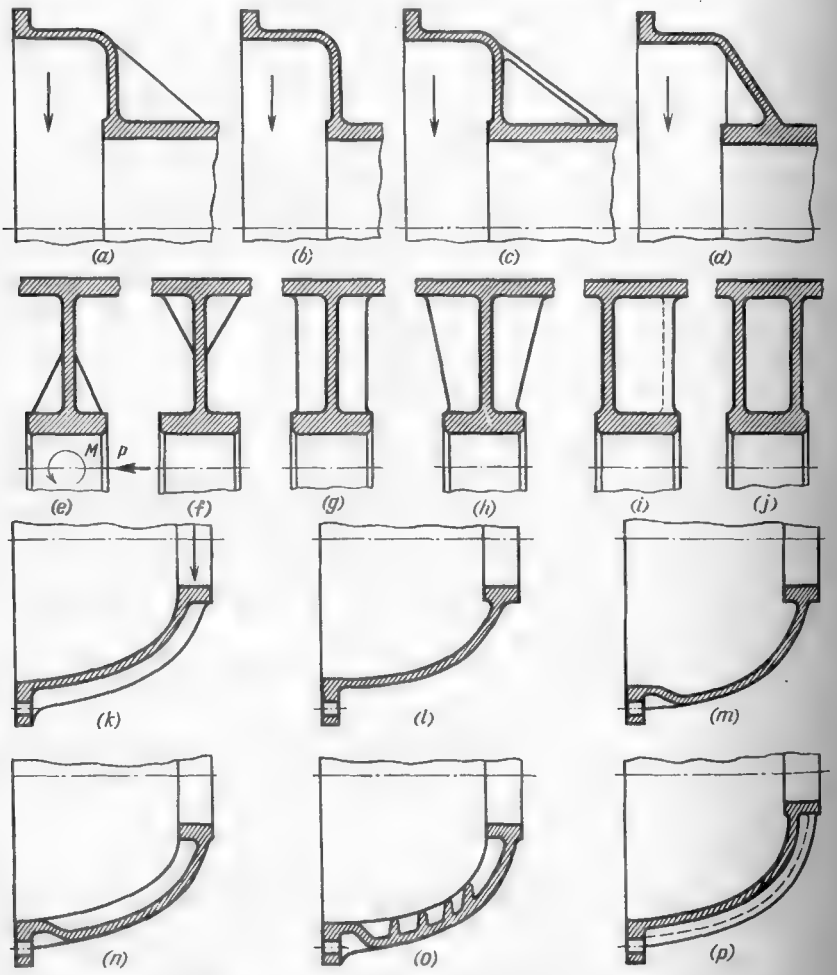


Fig. 150. Influence of rib forms and position on component strength

Structural Examples

Figure 150 shows examples of correct and incorrect rib design.

A housing-type component with a rib working in tension at the transition point of two sections (Fig. 150*a*) is not at all advantageous to strength. Removal of the rib increases the housing strength (Fig. 150*b*). When introducing a rib, it is better to make it T-shaped (Fig. 150*c*) or position it so that the rib works in compression (Fig. 150*d*).

Figure 150*e-j* illustrates compartments of cylindrical housing with a partition (diaphragm), loaded with a transversal force P or bending moment M. The short ribs (Fig. 150*e*, *f*) weaken the partition.

The best constructions have ribs of constant height (Fig. 150*g*) or ribs which widen to the place of fixing (Fig. 150*h*).

The highest strength is possessed not by ribbed components but by ones which have fluted partitions (Fig. 150*i*), or box sections (Fig. 150*j*), especially when reinforced with inner transversal ribs.

Depicted in Fig. 150*k-p* are spherical cantilevered housings. Sometimes such components are ribbed on the outside (Fig. 150*k*). If the rib height is not great in relation to the wall thickness, the ribbing weakens the component. Removal of the ribs (Fig. 150*l*) enhances component strength. Still stronger components are obtained when their walls are enlarged within the limits of the overall sizes (Fig. 150*m*). Even farther increase in strength may be obtained by providing the component with internal longitudinal (Fig. 150*n*) or wafer-like (Fig. 150*o*) ribs. High rigidity and strength are possessed by components with fluted walls (Fig. 150*p*).

4.4. Improving the Rigidity of Machine Constructions

Ways of improving rigidity and strength of typical engineering structures are given in Table 20.

(*a*) *Fixing of Cantilevers*

The rigidity of cantilevered systems is strongly affected by the method of fixing. Constructional measures can give a cantilever of any rigidity, but these measures may be brought to nought if the cantilever fixing is weak or if the cantilever is set into a non-rigid component. Ways of improving the rigidity of cantilevered systems are given in Table 21.

Table 20

Rigidity Improvements of Engineering Structures

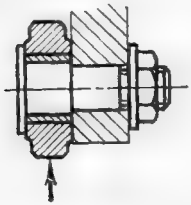
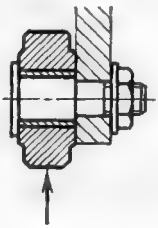
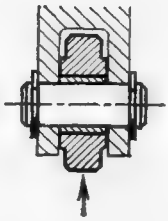
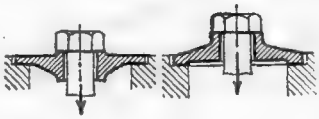
Original design	Improvement	Essence of improvement
Cantilever fastening of a roller lever 		Cantilever span is reduced. Lever, shaft and assembly strengthened
Lever weak, roller shaft fastening non-rigid		Cantilever eliminated; roller shaft mounted in two supports in lever fork
Flat bearing washer loaded in flexure by axial force Poor rigidity		Reinforced by circular collar
		Wasner body given an equal resistant form
		Washer given conical form; flexural stresses sharply decreased.

Table 20 (continued)

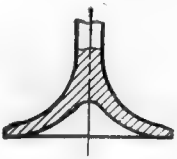
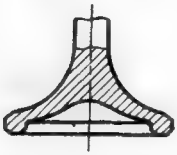
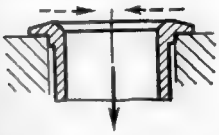
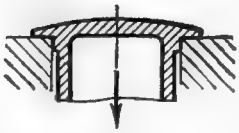
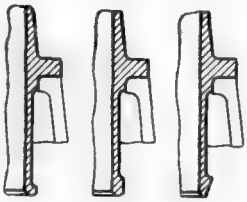
Original design	Improvement	Essence of improvement
Internal combustion engine valve		Valve plate given tulip-shaped form. Plate rim rigidity still poor
Non-rigid valve plate; plate-to-stem link poor		Stem and plate made more solid; rigidity of plate rim improved
Tubular part loaded by axial force 		Rim-to-cylinder joint strengthened
Support rim flexing under load converges towards centre (dashed arrows). Ultimately part may be forced through hole		Shoulder-to-shoulder web is provided (most rigid design)
Skirt of piston engine cylinder Skirt deforms under transverse piston load		Annular stiffening ribs at skirt bottom end

Table 20 (continued)

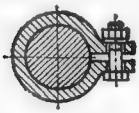
Original design	Improvement	Essence of improvement
Sleeve loaded by transverse force Under loading, sleeve edges loose their cylindrical shape		Sleeve edges strengthened by bead
Terminal-connection Tightening of terminal deforms shaft journal		Journal deformation eliminated by introducing wed
Terminal connection  Terminal lugs bend when tightened. Power tightening not possible		Terminal strengthened. Coupling bolt positioned closer to shaft

Table 20 (continued)

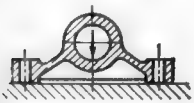
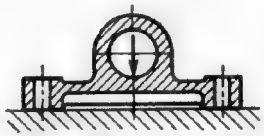
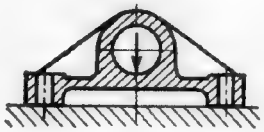
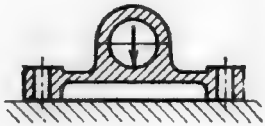
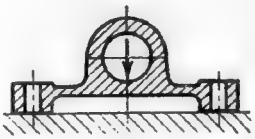
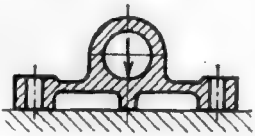
Original design	Improvement	Essence of improvement
Cast lug  Arch of base is subjected to heavy flexure		Arch of base reinforced by internal ribs, working in tension
		Arch of base reinforced by external ribs, working in compression
		Arch of base given rigid pyramid form
		Arch of base provided with peripheral frame; frame rests on seating surface
		Lug provided in acting load plane which takes the load (most light weight construction)

Table 20 (*continued*)

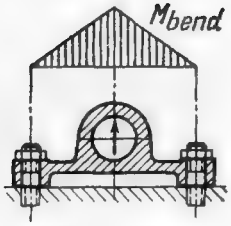
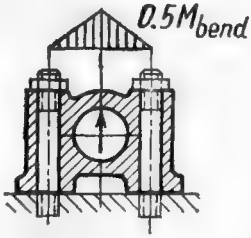
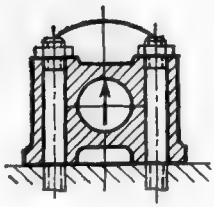
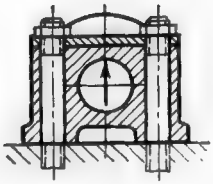
Original design	Improvement	Essence of improvement
Bearing  Fastening bolts spaced wide by apart; bending moment at critical section has large value	 	Bending moment reduced by closer fastening
		Moment of resistance and inertia at critical section increased by rib
		Critical section is reinforced by steel strap (such method applied to light alloy bearings)

Table 20 (continued)

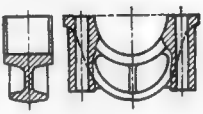
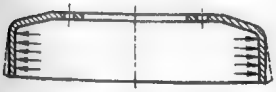
Original design	Improvement	Essence of improvement
Suspended bearing Rigidity poor		Section strengthened
		Rib added (constructions are in order of increased rigidity)
		Section given I-beam form
		Section made box-shaped (best design for strength and rigidity)
Rim of brake drum  Rim deforms under action of brake shoes; brake shoes work only at edges		Stiffening flange added

Table 20 (continued)

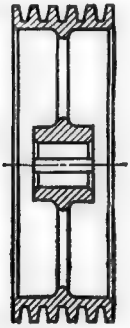
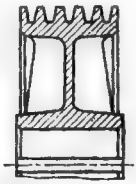
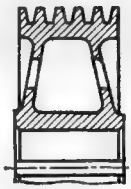
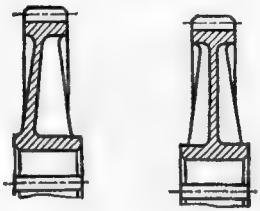
Original design	Improvement	Essence of improvement
Cast multigroove V-belt pulley  Spoke construction. Rigidity poor		Rim joined to hub by solid ribbed disk. Hub lengthened
		Section given conical box form (best design for rigidity)
Spur gear Rigidity poor		Disk of gear made conical
		Ribs added to spur gear disk (for cast gears)

Table 20 (continued)

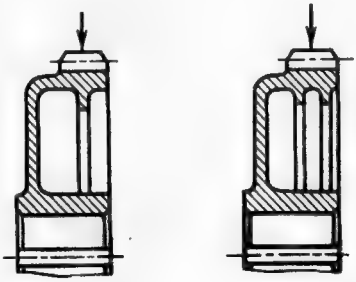
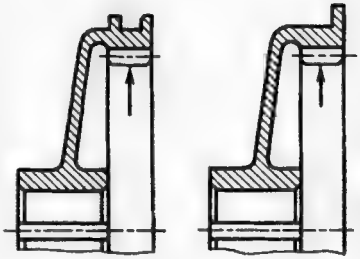
Original design	Improvement	Essence of improvement
External cup-shaped gear Driving forces deform rim		Internal stiffening rib(s) introduced
Internal cup-shaped gear Rigidity poor		External stiffening rib(s) introduced

Table 20 (continued)

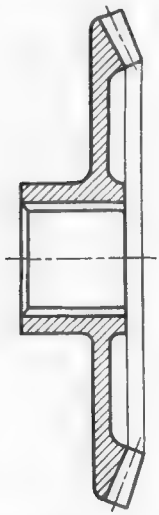
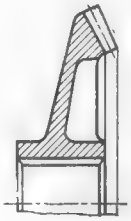
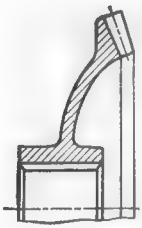
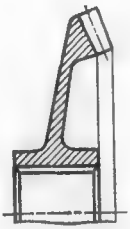
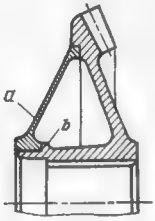
Original design	Improvement	Essence of improvement
Bevel gear  Rigidity poor		Disk made conical
		Disk made spherical
		Ribs added (for cast gears)
		Welded box-shaped prestressed construction. Clearance, between added body *a* and shoulder *b*, is eliminated before welding. Teeth and splines machined after welding

Table 20 (continued)

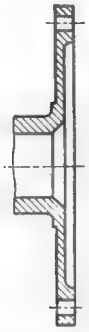
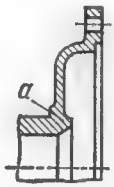
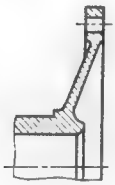
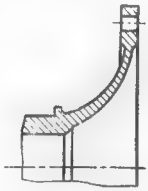
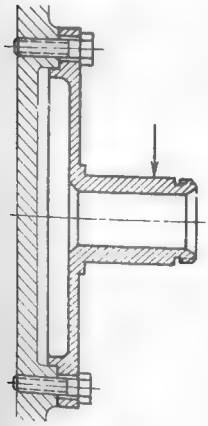
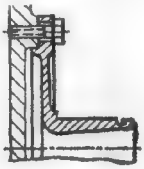
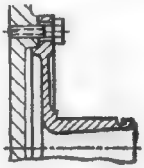
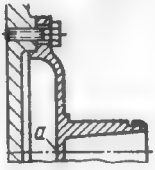
Original design	Improvement	Essence of improvement
Flanged shaft  Rigidity poor		Flange made cup-shaped. Transition section *a*, where flange joins hub, strengthened
		Flange made conical
		Flange given tulip-form. Discontinuity of force flow eliminated
Bracket with trunnion  Rigidity poor		Disk and trunnion made conical
		Disk made cup-shaped. Web *a* provided, which strengthens transition section
		Disk given better conical form

Table 20 (continued)

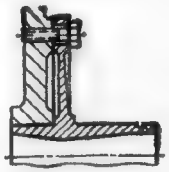
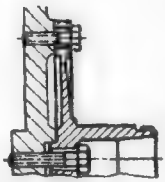
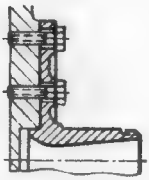
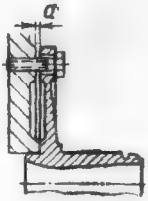
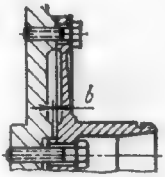
Original design	Improvement	Essence of improvement
		Trunnion lengthened and force-fitted into housing body
		Disk attached to housing body by additional central bolt
		Disk attached to housing body by two rows of radially spaced bolts
		Prestressed construction. Clearance *a* between disk and body eliminated by tightening
		Prestressed construction. Clearance *b* between trunnion and body eliminated when tightening central bolt

Table 20 (continued)

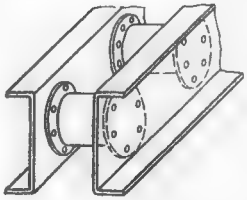
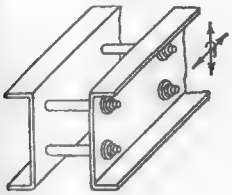
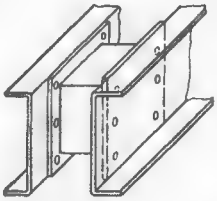
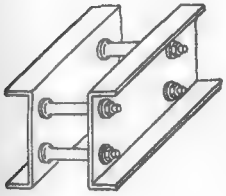
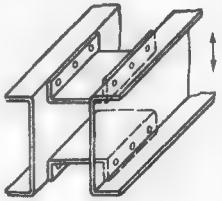
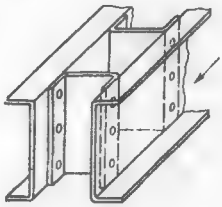
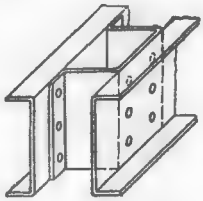
Original design	Improvement	Essence of improvement
		Connected by box-pattern members (box-pattern members are expensive to manufacture)
Composite beam, built from two thin-walled U-sections (direction of work loads shown by arrows) 		
 Rigidity poor		Connected by bent sections (transverse rigidity not assured)
		Connected by bent sections (longitudinal rigidity not assured)
		Connected by U-sections (longitudinal rigidity not assured)

Table 20 (continued)

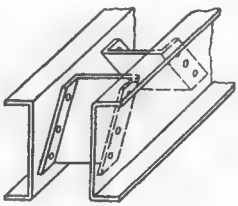
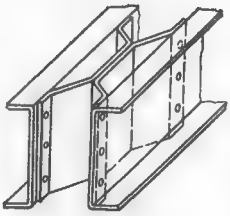
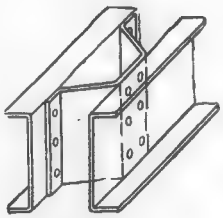
Original design	Improvement	Essence of improvement
		Diagonal connections (rigidity assured in all directions)
		Trapezoidal connections (rigidity assured in all directions)
		Trapezoidal connections arranged alternately alond the beam. Best constriection in terms of rigidity, weight and manufacture

Table 20 (continued)

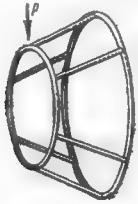
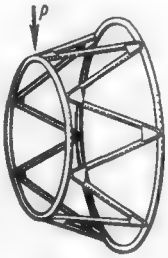
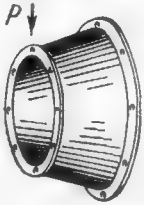
Original design	Improvement	Essence of improvement
Frame bracket loaded by force P  Rods work mainly in flexure. Stresses in the system very high. Rigidity poor	 	Construction given a truss form, rods arranged in triangular diagonal pattern. Rods work mostly in tension-compression When loaded by a transverse force, the side rods, positioned parallel to the plane of acting bending moment, accept the load; the rods, positioned perpendicularly to that plane also participate in the work as space trusses, whose side rods serve as bracing struts. The truss form is closed by the front stiffening ring To give the system full determinancy, the ring-to-rod joints are articulated
	 	Shell-type construction. A highly rigid construction. Weight can be reduced by cut outs
	 	Shell lightened by box-shaped cut outs Non-rational design, since parts between cut outs work in flexure
	 	Shell construction given more rationally shaped cut outs

Table 20 (continued)

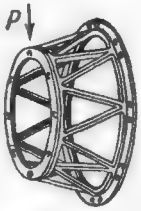
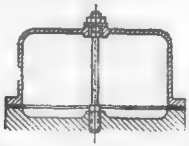
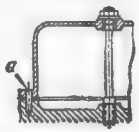
Original design	Improvement	Essence of improvement
		Weight-relieved construction
		Moulded truss construction
Cast cover Rigidity poor		Cover ribbed for rigidity
		Cover made vault-shaped
		Cover made pyramid-shaped
Box-shaped cover fixed to housing by central studs  Rigidity poor. Cover deflects when tightened down		Side deformation arrested. Cover positioned by check pins *a*, which wrestle distortion of vertical walls (horizontal wall is still non-rigid)

Table 20 (continued)

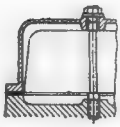
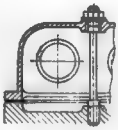
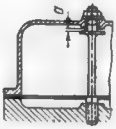
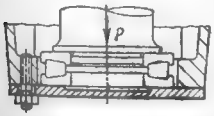
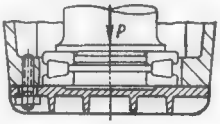
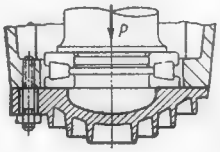
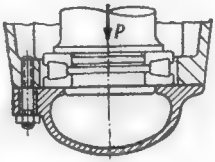
Original design	Improvement	Essence of improvement
		Internal ribs added
		Internal partitions added
		Cover made vault-shaped
		Deformation restricted (degree of deformation determined by initial clearance *a* between the cover and limiting stop shoulder of stud)
Loaded cover, accepting the load from a vertical footstep shaft bearing  Poor rigidity and strength		External circular and radial stiffening ribs provided
		Circular ribs provided and cover made vault-shaped
		Cover given arched form

Table 20 (continued)

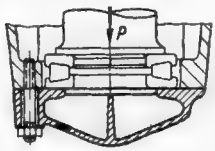
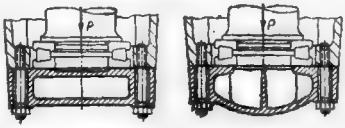
Original design	Improvement	Essence of improvement
		Arched form and internal ribs
		Cover given box form (very rigid and strong design)
Diaphragm supporting plain bearing Rigidity poor		Diaphragm ribbed
		Diaphragm made conical ribbed
		Given cup-shaped form
		Fluted

Table 20 (continued)

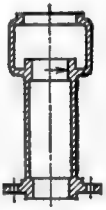
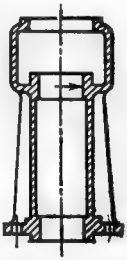
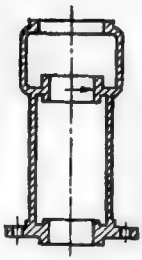
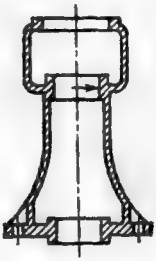
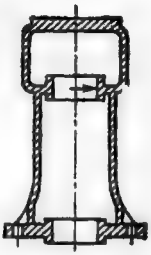
Original design	Improvement	Essence of improvement
Cast bracket, loaded in flexure  Rigidity poor		Ribbed bracket column
		Radial size of bracket column increased
		Column made conical. Connection to fastening flange strengthened
		Radial size of bracket column increased. Column-to-flange joint made conical

Table 20 (continued)

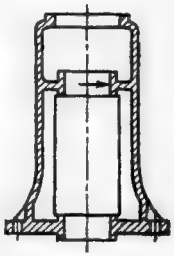
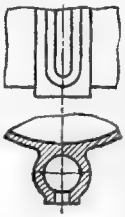
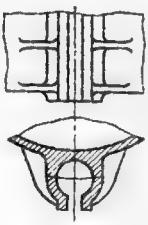
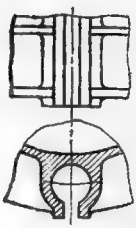
Original design	Improvement	Essence of improvement
		Radial size of bracket increased to maximum overall size. Internal ribbing introduced (construction very rigid and strong)
Cast revolving drum. Reciprocating along cylindrical guides are operating rods with rollers, which roll on a stationary templet as the drum revolves		Non-through slot. Design not technological. Very difficult to assemble rod and roller unit
		Lugs reinforced by external ribs
The through slot strongly weakens the lugs. Under action of operating forces the lugs walls move apart (see arrows), thus spoiling rod directional movement		Lugs reinforced by continuous circular ribs

Table 20 (continued)

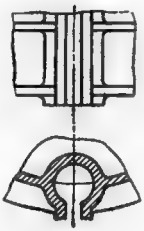
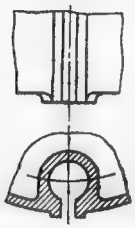
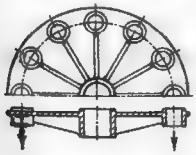
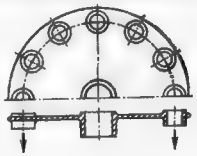
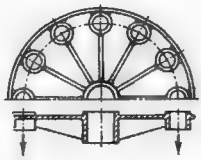
Original design	Improvement	Essence of improvement
		Radial drum size increased. Lugs strengthened by ribs
		Radial drum size increased to maximum size (best rigid and strong construction)
Cast turntable of rotary machine; loaded by bending forces acting at the operative sections  Rigidity poor	 	Ribs linking the central hub to the peripheral bosses are introduced Peripheral rigidity is enhanced further by circular rib

Table 20 (continued)

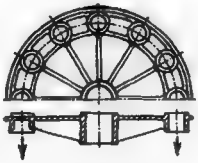
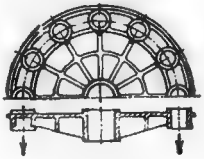
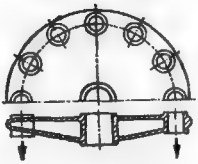
Original design	Improvement	Essence of improvement
		Peripheral bosses hoop still further strengthened by second circular rib
		Turntable rigidity improved by radial and circular interconnected system of ribbing
		Turntable given box shape (most rigid construction)

Table 21

Improving Rigidity of Cantilevered Systems

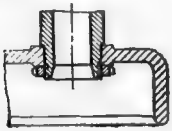
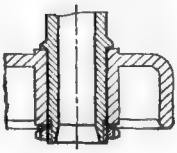
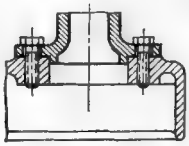
Original design	Improvement	Essence of improvement
Installation of cylindrical column into a cast housing  Rigidity poor		Axial fixing. Extended end of column fixed in rigid housing boss
		Radial fixing. Column given a flange and attached to strengthened housing surface

Table 21 (continued)

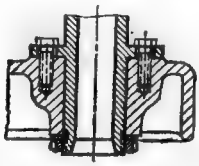
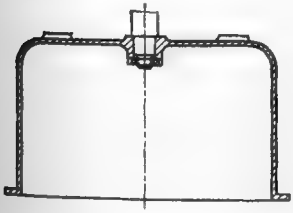
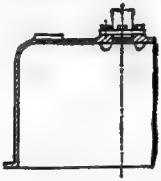
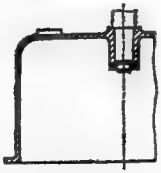
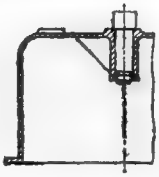
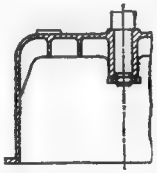
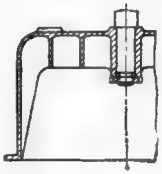
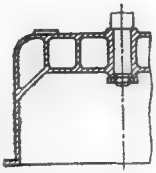
Original design	Improvement	Essence of improvement
		Radial and axial fixing
Attachment by welding rod to U-shaped channel Rod unstable in transverse direction		Rod fixed to both channel flanges
		Channel flanges strengthened by webs where the rod is welded to channel
Fixing a column to a cast housing  Column is unstable owing to compliance of housing top		Fixing strengthened radially (housing top still remains compliant)
		Fixing strengthened axially (housing top still remains compliant)

Table 21 (continued)

Original design	Improvement	Essence of improvement
		Local internal ribbing which only strengthens central part of top
		Extended internal ribbing which strengthens entire top
		Ribbing extended further, strengthens corners, top and part of vertical walls
		Ribbing extended still further, walls strengthened down to base
		Webbing takes box form (the most rigid construction)

(*b*) *Column Bases*

Cast steel footings joined with the column body by welding (Fig. 151*a*) impart high rigidity and strength to the entire support. However, such constructions are very complex to manufacture and heavy.

In the design shown in Fig. 151*b* the foot is a plate welded to the column end. The rigidity of the joint is insufficient. Such construc-

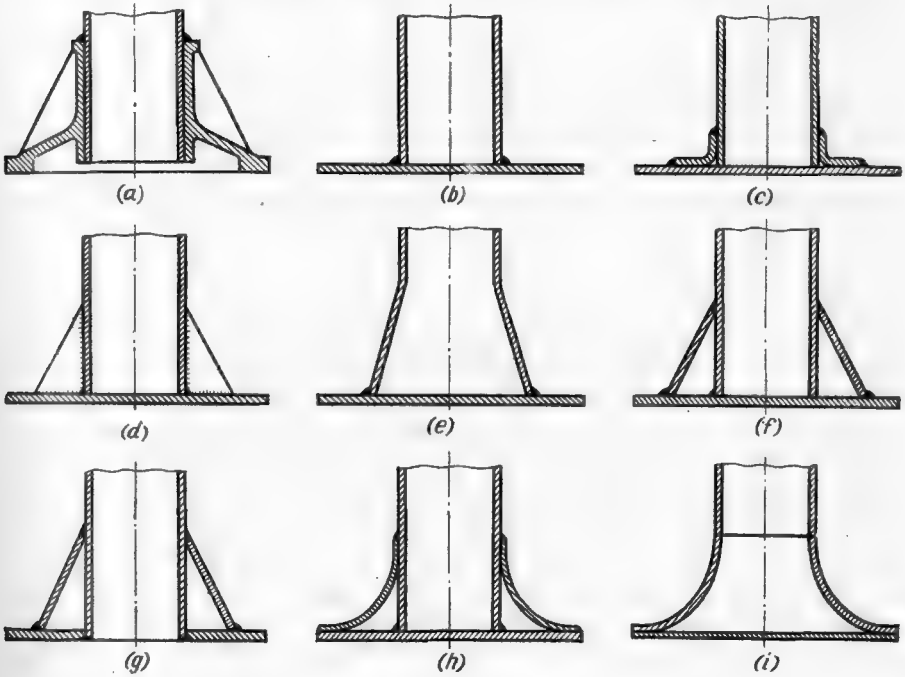


Fig. 151. Column bases

tions are suitable for light props subjected to moderate loads. The rigidity of column fixing can be increased further by the addition of a welded formed collar (Fig. 151*c*) or gussets (Fig. 151*d*). The latter design is widely used as it is simple to manufacture and sufficiently rigid.

The column end (Fig. 151*e*) is flared when it is necessary to make the external appearance more attractive. With large column sizes flaring can be difficult.

More practical from the viewpoint of manufacture are welded-on cone constructions (Fig. 151*f*, *g*). The stiffening element is often made in the form of a streamlined torus (Fig. 151*h*).

A tulip-shaped base butt-welded to the column walls has the highest rigidity (and best outward appearance) as can be seen in Fig. 151*i*.

(*c*) *Rigidity of Housing Components*

The main ways to improve the rigidity of housing components without increasing their weight (sometimes even with lowering it) are: rounding of transition points, making walls vault shaped, efficient (internal) ribbing, introduction of interwall struts (preferably diagonal).

The rigidity of housings can also be improved constructionally uniting housing elements (monoblock constructions).

Figure 152 shows (approximately in the order of historical succession) constructional strengthening of in-line internal combustion engines. In an engine with separate cylinders (Fig. 152*a*) the rigidity is determined only by the rigidity of the crankcase. With the flexure forces occurring during ignition, the crankcase is deformed and so is the entire engine.

More rigid is the half-block construction (Fig. 152*b*) where the cylinder heads are united into one general block. The summed moment of inertia of the system, strengthened by a cylinder head block and overall camshaft cover, sharply increases.

The systems having wide use and preferred in the motor industry are the block systems (Fig. 152*c*, *d*). Here rigidity is enhanced by making the cylinder jackets in one common block which is either attached to the crankcase (Fig. 152*c*) or cast integrally with it (Fig. 152*d*). The latter alternative gives the most rigid and strongest design with the smallest number of joints between elements.

Of no less importance is the rigidity of the crankcase incorporating crankshaft supports, since the former is subjected to flexure in the plane of acting ignition forces, i.e., in the longitudinal plane of crankcase symmetry. To increase rigidity, it is advisable to improve the moments of inertia across the transverse sections of the crankcase and prevent the side walls from "opening" by providing stiffening transverse webs between the walls.

Figure 153 shows simplified constructional examples of crankcase designs used with detachable cylinder jacket units (ref. to Fig. 152*c*).

The system, presented in Fig. 153*a*, comprises the main crankcase (top half, depicted in heavy lines) and sump, and has rather poor rigidity, though it is very convenient for installation and assembly of the crankshaft.

The split plane of the main crankcase and sump lies above the shaft axis. The shaft is held in suspension bearing *1*.

The main crankcase can be made very robust by shifting the split plane to the crankshaft axis and diminishing accordingly the sump height (Fig. 153*b*). In the design illustrated in Fig. 153*c*, the crank-

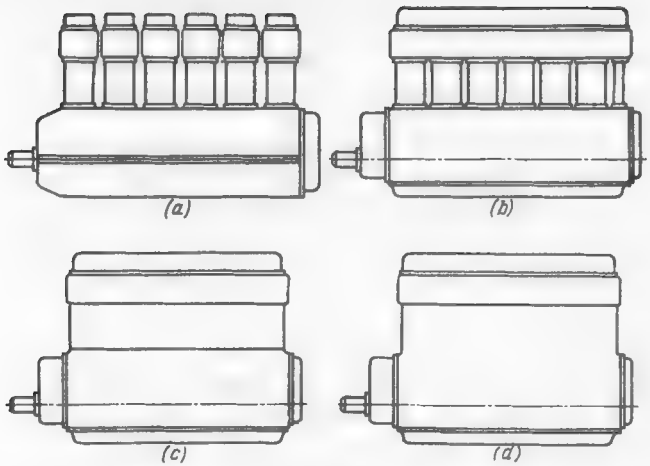


Fig. 152. Constructional evolution of internal combustion engines

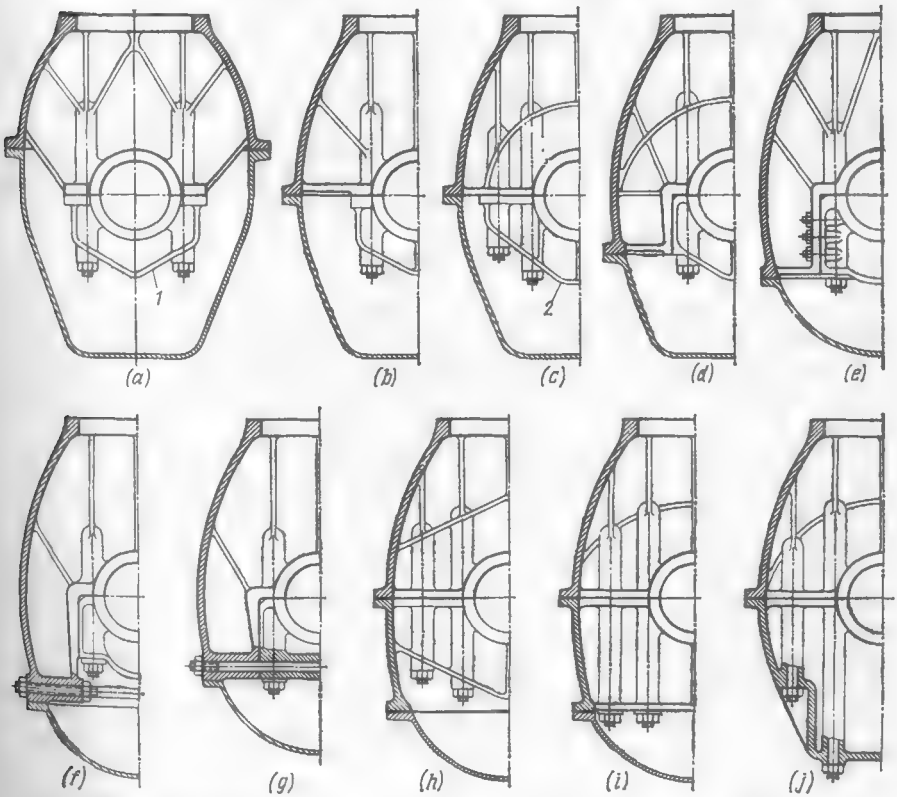


Fig. 153. Crankcase constructions of in-line internal combustion engines

case transverse partitions are stiffened with arched ribs. Bearing suspensions *2* have been developed in the transverse direction and secured to the case by two rows of bolts, as a result of which rigidity points are formed around the crankshaft supports.

The rigidity of the main crankcase can be further improved by shifting the split plane below the shaft axis (Fig. 153*d*). To strengthen the bond between the crankcase side walls, the arched ribs are extended to the walls and the bearing suspensions are mounted in the crankcase cut-outs.

Wall-to-wall strength is enhanced with the aid of bolts which join bearing suspensions and crankcase partitions together (Fig. 153*e*). In the design pictured in Fig. 153*f* the crankcase walls are joined by tie-bolts; to avoid overtightening, the value of the internal free span is adjusted by nuts.

A more rational construction is the one in which the tie-bolts are tightened on to the suspension wall support (Fig. 153*g*).

Further improvements in the rigidity of the system can be obtained by making the main crankcase in two parts with the split line plane along the shaft axis (Fig. 153*h*, *i*). The suspended bearings in this case are integral with the lower crankcase half.

The most rigid design is presented in Fig. 153*j*: the crankcase is comprised of two parts (halves) connected together at the shaft axis plane by two rows of clamping pins. Both halves are load-carrying elements and equally take part in flexure.

(*d*) *Plates*

Figure 154 illustrates methods of improving rigidity and strength of moulded plates. It is assumed that the plate is loaded in the centre and rests upon four side supports (at the corners).

The original design (Fig. 154*a*) has poor rigidity and strength. When provided with longitudinal ribs which offer uniform resistance to flexure (Fig. 154*b*), a higher longitudinal rigidity is obtained, but transverse rigidity remains inadequate.

Equal rigidity in the longitudinal and transverse directions is obtained when the plate is furnished with radial ribs (Fig. 154*c*).

Another design principle has been embodied into the alternative presented in Fig. 154*d*, the rigidity of which is improved by the addition of vertical walls around the plate perimeter. Flexure deformations are restrained by the walls which work in tension. The rigidity can be enhanced still further by increasing the wall height, by increasing edge sections and connecting the plate body to the walls by ribs (Fig. 154*e*) which will transfer the flexure deformations to the edge walls.

Drawing the side walls together by tie-bolts (Fig. 154*f*) create stresses in the plane of opposite sign to those of the working stresses

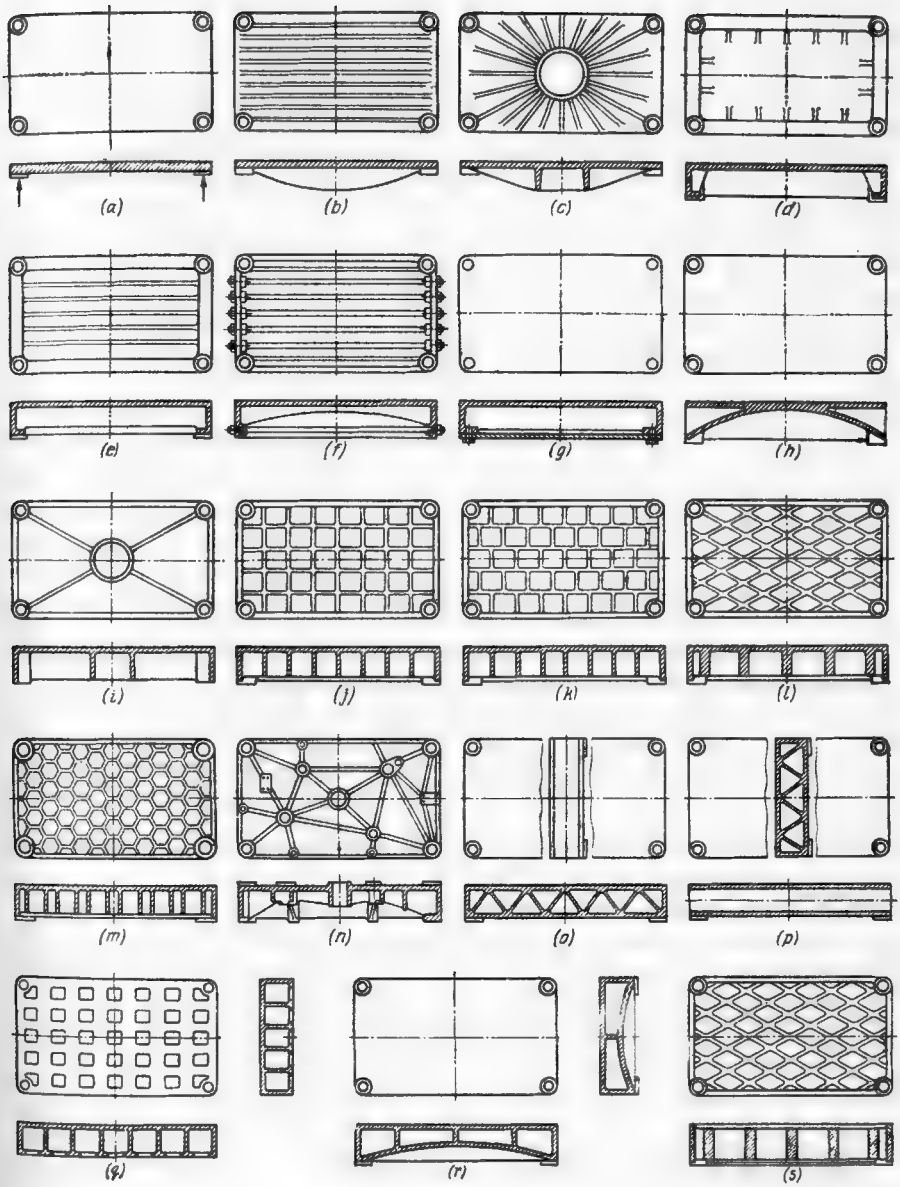


Fig. 154. Design of cast plates

High rigidity and strength are possessed by constructions having a sheet steel cover which works in tension (Fig. 154*g*). By heating the cover prior to mounting prestressed conditions can be created if it is positively secured to the plate (e.g., by means of locating pins).

Another way of imprqving rigidity is by making the plate vault-shaped (Fig. 154*h*). Side walls with diagonal (Fig. 154*i*), wafer-like (Fig. 154*j*), checkered (Fig. 154*k*), diamond (Fig. 154*l*) and honeycomb (Fig. 154*m*) ribbed constructions have high rigidity.

When fastening points on the plate are present, the ribs must be arranged so as to connect the centres of rigidity (Fig. 154*n*).

The highest rigidity is given by double-walled plates when internal slanted ribs (Fig. 154*o*, *p*) formed during casting by means of through cores secured on prints in the plate side walls.

Close to these are half-enclosed plates with internal cells moulded during casting by mould core assemblies secured on prints through holes in the plate bottom (Fig. 154*q*) and double-walled plates with concave bottoms (Fig. 154*r*).

Figure 154*s* illustrates a light easily producible design of lattice type. For a smooth external surface such plates are usually faced with thin-sheet metal skins.

(*e*) *Rigidity of Thin-Walled Constructions*

In constructions made from sheet materials (shells, thin-walled sections, reservoirs, skins, panels, covers, etc.) it is necessary to consider not only the deformations caused by working forces but also the strains arising during welding, machining, joining and tightening of assembled elements. The possibility of chance damage to walls during transit, installation and carelessness in use must also be considered.

In loaded shell constructions prevention of stability losses is of primary importance.

The chief methods of increasing rigidity remain the same: maximum relief of flexure, changing flexure stresses to tensile-compressive ones, introducing ties between planes of maximum deformation, enlarging sections and moments of inertia at critical points, employing reinforcing elements at places where loads are concentrated and at parts of discontinuous force flows, using conical and arched forms.

Compartments. The radial rigidity of large size thin-walled cylindrical components (compartment-like units) is enhanced by means of annular stiffening collars (Fig. 155*a-i*).

Compartment constructions with double walls are more rigid and strong (Fig. 156*a*). To improve radial rigidity, it is advisable

for compartment walls to be joined with each other. Sometimes one restricts himself to introducing local ties, i.e., welding strengtheners to compartment walls (Fig. 156*b*) or by welding tubes into them (Fig. 156*c*).

The best results are obtained when annular stiffening collars are used (Fig. 156*d-g*). No less efficient is subdividing the entire compartment into several shorter subcompartments (Fig. 156*h*, *i*). In the latter case the role of stiffening collars is fulfilled by subcompartment butt-joints.

Introduction of cones (Fig. 156*j*) or arched elements (Fig. 156*k*, *l*) increases not only radial but also longitudinal rigidity.

Figure 157*a-c* presents satisfactory compartment designs, reinforced by conical elements.

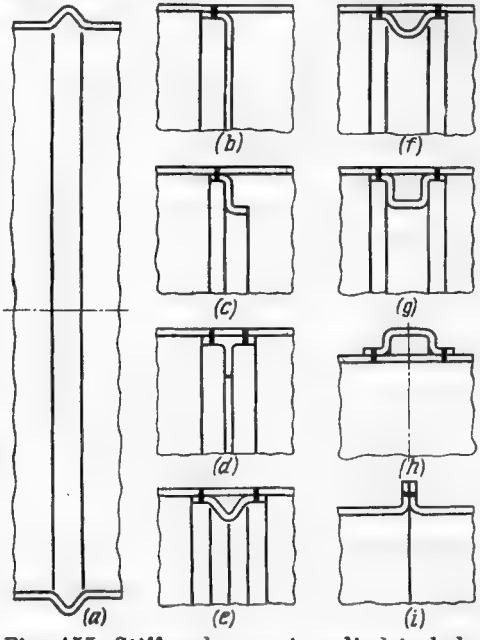


Fig. 155. Stiffened zones in cylindrical shell components

Longitudinal rigidity of compartments is attained by ribs positioned along the cylinder generatrices (Fig. 158*b-g*). Components

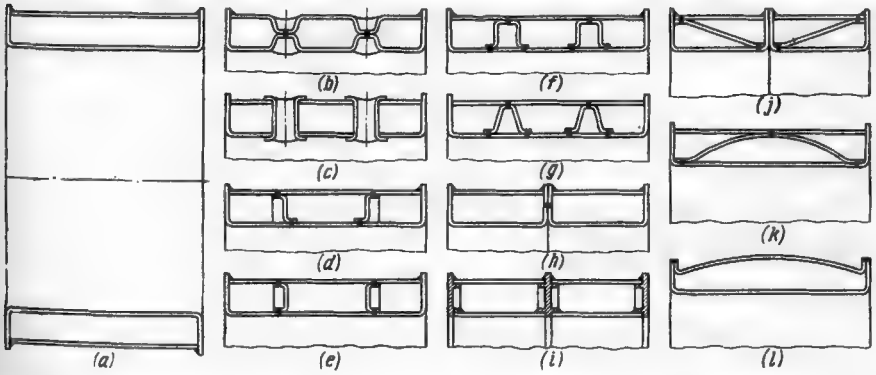


Fig. 156. Enhancing the radial rigidity of double-walled compartments

of the highest rigidity are obtained when combining longitudinal ribs and stiffening rings (Fig. 158*a*).

Helical and zigzag ribs (Fig. 159) better torsional rigidity; their manufacture, however, is more difficult than that of straight longitudinal ribs.

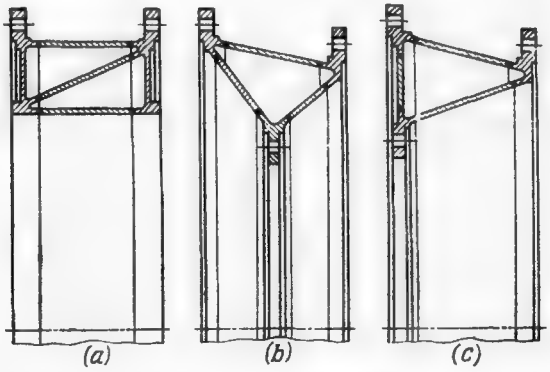


Fig. 157. Compartments reinforced with conical elements

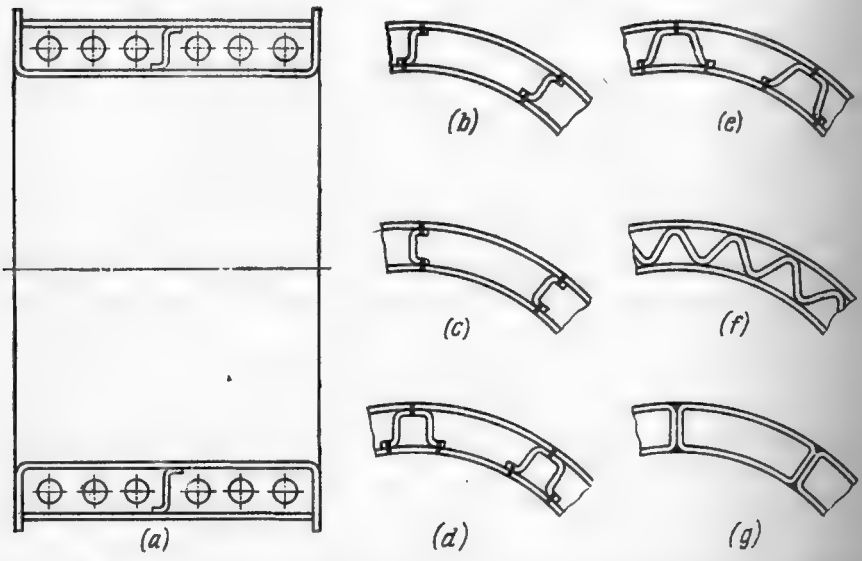


Fig. 158. Enhancing the longitudinal rigidity of double-walled compartments

Double compartments are joined together by means of external (Fig. 160*a-c*) and internal (Fig. 160*d*) flanges. The latter contribute to still higher rigidity and substantially decrease the radial dimensions of components.

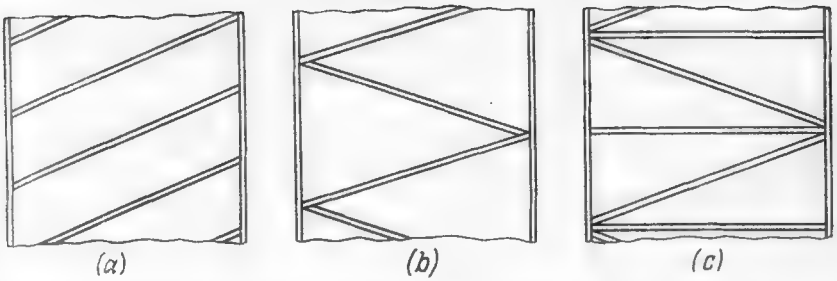


Fig. 159. Increasing the rigidity of shell systems with diagonal ribs working in torsion

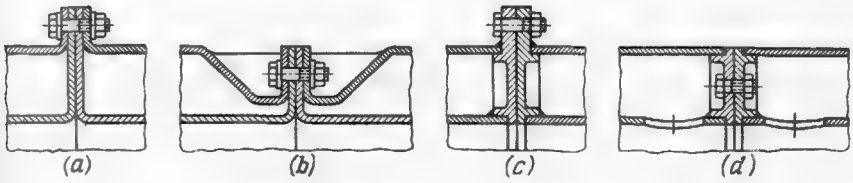


Fig. 160. Methods of joining shell compartments

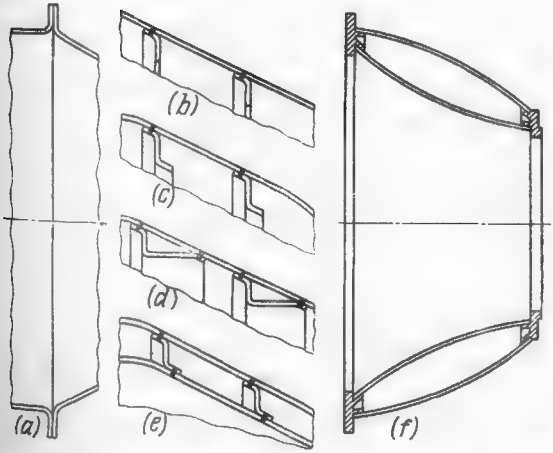


Fig. 161. Enhancing the radial rigidity of conical shell systems

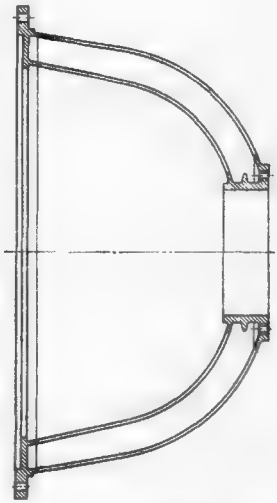


Fig. 162. Spherical cantilevered double-walled member

When planning the introduction of bolts from inside it is necessary to provide holes in the internal walls large enough to introduce and tighten the bolts.

Figure 161*a-f* illustrates enhancement of radial rigidity in conical compartments; Fig. 162 shows the construction of a double-walled spherical bracket-type component.

Shell constructions with spatial lattices. The highest rigidity of a shell-system can be obtained by filling the shell interior space with uniformly distributed stiffening elements connecting all their sections and turning the system into a spatial lattice working as one unit. The appearance of strong synthetic resins and adhesives allows this problem to be solved rather closely.

There are two basic kinds of spatial reinforced shells in use: foam plastic and honeycomb constructions.

In the first case the space between the metallic shells is filled with foamable plastics based on thermo-setting or curable resins. The plastics are introduced in a liquid state, with the addition of gassing agents and emulsifiers. When heated to 150-200 °C the plastic composition foams and hardens, forming a spongy mass up to 80-90% in volume and with specific weight $\gamma = 0.1\text{-}0.2$ kgf/dm^3.

As a result, the strength, rigidity and stability of the reinforced system considerably increase though not so much as when metallic spatial ties are introduced. The system is generally applied in conjunction with metallic stiffening elements, either transverse (ribs, frames) or longitudinal (spars, stringers).

Honeycomb constructions are prepared by joining together honeycomb-pattern impressed fabric or glass-fibre cloths impregnated with thermo-setting or curable resins. Mantles are made of sheets of the same material or of metal sheets. The size of honeycomb cells is usually 8 to 15 mm.

Still better in strength and rigidity are metallic honeycombs made by gluing together embossed metal sheets covered with a film of phenol-neoprene adhesives or modified epoxides. The same adhesives serve simultaneously for joining metallic mantles to honeycomb structures whose strength is dependent on the adhesion (the resistance of strongest synthetic adhesives to shear comprises 2-5 kgf/mm^2, and bond strength 5-10 kgf/mm^2).

Steel sheets can be joined more strongly by furnace brazing with bronze alloys in vacuum or in a reducing atmosphere.

New possibilities for making strong honeycomb structures are opened by fine-focus electron-beam welding technique. The welding temperature is produced only at the focus point, the remaining zones do not undergo significant heating. This enables butt-welding to be done at any depth with one and the same position or attitude of the welding unit. Depth adjustment in welding is effected by refocusing the electron beam with the aid of converging magnetic

coils and in the transverse and longitudinal directions by means of deflecting coils. Thus, all inner joints in the component can be successively welded.

Stability of shell constructions. The increase in overall dimensions and reduced wall thicknesses in the first instance mean enhancing radial rigidity and elimination of component stability losses from loads.

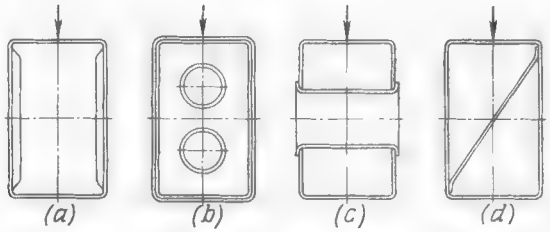


Fig. 163. Rectangular thin-walled beams with transverse ties

The rigidity of welded thin-wall square-section beams is improved by embossing reliefs (preferably oblique) on the walls (Fig. 163*a*), introducing transverse partitions (Fig. 163*b*), providing tubular or box-section connection elements (Fig. 163*c*), and diagonal stringers (Fig. 163*d*).

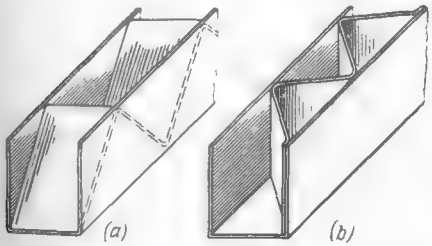


Fig. 164. Thin-walled beams with transverse struts

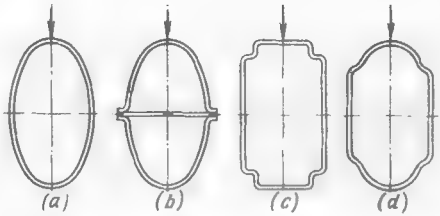


Fig. 165. Cross-sections of thin-walled high-rigidity beams

High rigidity is ensured by serpentine-like diagonal partitions (Fig. 164).

Figure 165 illustrates cross sections of beams, possessing high rigidity and stability.

Reinforcement of areas subject to concentrated loads. In the design of thin-walled components particular attention should be paid to areas subjected to concentrated loads. Insufficient rigidity at these parts causes local wall deformations making the construction unfit for use.

For cylindrical shell parts the simplest procedure is to introduce straps which distribute the force over a larger area (Fig. 166*a*, *b*). Still more efficient are stiffening collars and partitions (Fig. 166*c-e*) distributing the force over the entire component cross section.

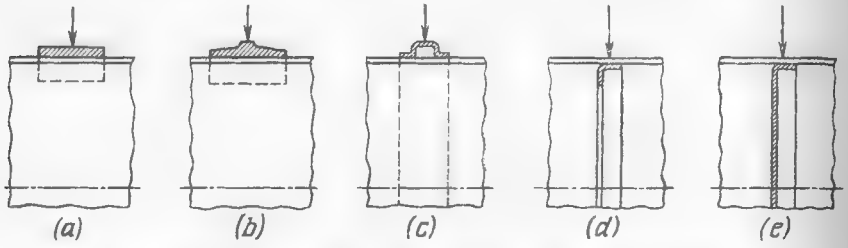


Fig. 166. Reinforcement of shell structures at load concentration points

The flexure of thin-walled components in bolt fastening areas (Fig. 167*a*) is prevented by using large-diameter washers (Fig. 167*b*), flanging the walls (Fig. 167*c*, *d*) or introducing some stiffening elements (Fig. 167*e*, *f*). The most efficient technique is the taking

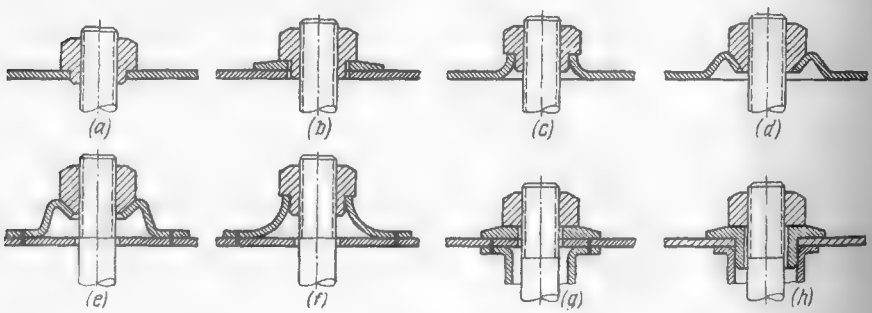


Fig. 167. Reinforcement of fastenings

up of the lightening forces by a strut-like element (e.g., a tubular column) working in compression (Fig. 167*g*, *h*).

Figure 168 shows how a thin-walled cover is attached to the main component by means of a non-loosable bolt. In the original design (Fig. 168*a*) the cover deforms even under light tightening pressure. To avoid sag, the tightening is limited to the preset gap *m* (Fig. 168*b-d*).

In the design presented in Fig. 168*d* the limiting element is made in the form of a conical catcher easing the introduction of the threaded bolt end when mounting the cover. The spring keeps the bolt straight when removing the cover and facilitates reassembly.

Butt joints of sheet structures. The rigidity of butt joints in thin-walled components is particularly important when the joints must be hermetic.

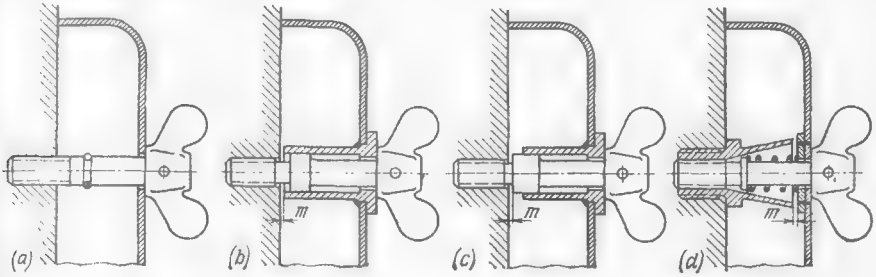


Fig. 168. Fastening of a thin-walled cover to a housing

When connecting flanges of two thin-walled cylindrical components of large diameter (Fig. 169*a*) no hermetic sealing can be effected in sections between bolts because of the insufficient rigidity of the

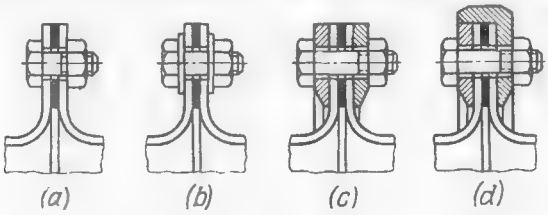


Fig. 169. Joints of thin-walled cylindrical parts

flanges. Washers placed under the bolt heads and nuts hardly improve rigidity (Fig. 169*b*). It is possible to make the joint tight by introducing additional solid rings, clamped or welded in position (Fig. 169*c*, *d*).

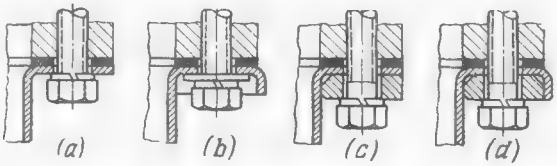


Fig. 170. Fastening of a formed sheet steel sump to a housing

If a stamped sheet steel sump is to be secured to a housing (Fig. 170*a*), tight sealing is accomplished by bending the flange (Fig. 170*b*) or by reinforcing with a solid frame tack-welded to the sump (Fig. 170*c*, *d*).

Stiffening reliefs. Rigidity can be enhanced by embossing reliefs in walls (Fig. 171), generally in the form of beads. To simplify the cold stamping process, the height of reliefs should not be greater

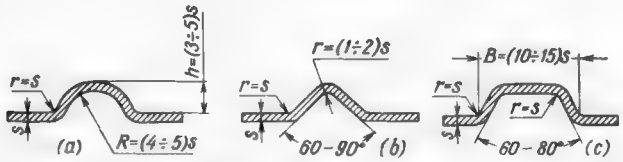


Fig. 171. Design forms of stiffening reliefs

than (3-5)s, where s = thickness of material. Reliefs of larger heights are stamped in several passes with intermediate annealing, this making the production more expensive.

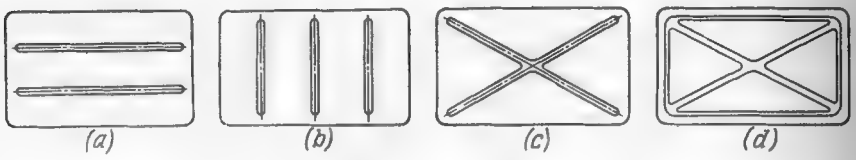


Fig. 172. Various relief forms on a rectangular cover

In hot-stamped designs higher and more extensive reliefs can be made.

Apart from better strength and rigidity due to purely geometrical relationships (increased moments of resistance and inertia of sections), cold embossed reliefs add to the toughness as a result of cold working.

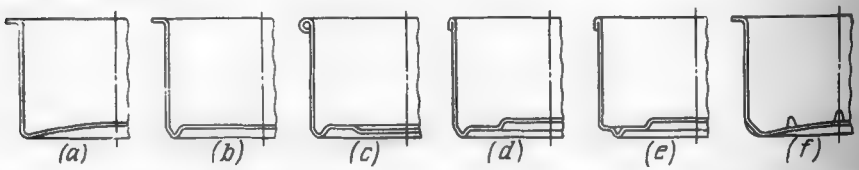


Fig. 173. Increasing the bottom rigidity of thin-walled cylindrical components

Relief beads should be positioned along the acting plane of the bending moment (Fig. 172*a*). The reverse position (Fig. 172*b*) does not improve rigidity, it only makes the component more compliant.

Reliefs must be directed toward the rigidity points of the system. For this reason, the best arrangement of stiffening beads for a square-shaped plate is a diagonal one (Fig. 172*c*, *d*).

Embossing the reliefs on the bottoms of cylindrical thin-walled vessels (Fig. 173) not only increases rigidity, but also improves stability and imparts definiteness to the vessel installation on a plane.

An effective way of increasing rigidity at angle transition points from edges to bottoms is by forming local triangular-shaped beads (Fig. 173*f*).

Figure 174 illustrates edge reinforcement of cylindrical reservoirs.

Fig. 174. Flanged edges on thin-walled cylindrical components

Lightening holes. Thin-walled constructions are often made with holes in order to reduce their weight. To increase local rigidity, to reduce stress concentrations and to enhance fatigue strength impaired by punching tools, the edges of the holes are reinforced by flanging (Fig. 175*a-c*), flanging with half- and full-curled edges (Fig. 175*d-f*), flanging with swaged edges (Fig. 175*g*, *h*) and also by reinforcing straps (Fig. 175*i-k*).

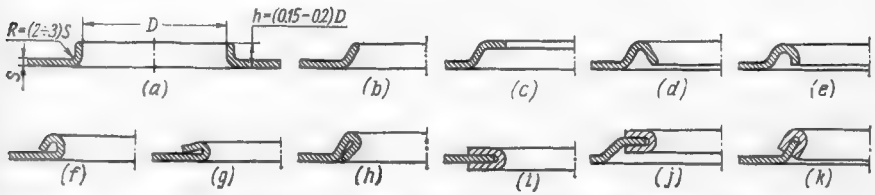


Fig. 175. Strengthening the edges of lightening holes

The flange height h (see Fig. 175*a*) should never be made too great since this entails greater manufacturing difficulties. The flange height attainable in a single cold flanging operation may be as great as $(0.15\text{-}0.25)D$. Higher flanges and also flanges with curled edges require several successive operations.

Another way to improve fatigue strength of material in the vicinity of holes is bilateral crimping of hole edges, using for the purpose dies and caulkers of rounded profile (Fig. 176).

Reservoirs. When designing reservoirs subjected to internal pressures it is necessary to avert wall bulging. Rectangular-section reservoirs (Fig. 177*a*) are not advantageous, because their walls bulge from the internal pressure (shown exaggerated by dash-lines). With such

shapes the addition of transverse stiffening webs is obligatory (Fig. 177*b*).

Better rigidity have oval, elliptic (Fig. 177*c-e*) and particularly cylindrical (Fig. 177*f*) reservoirs.

When reinforcing cylindrical reservoirs with external ribs, the direction of wall deformations must be considered.

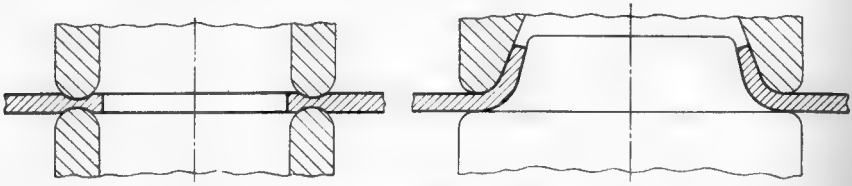


Fig. 176. Edge embossing

Tensile stresses in sections along generatrices

$$\sigma_1 = \frac{pD}{2s}$$

where p = internal pressure
D = diameter of reservoir
s = thickness of wall (Fig. 178*a*)

Stresses in cross-sections

$$\sigma_2 = p\frac{\pi D^2}{4\pi Ds} = \frac{pD}{4s} = 0.5\sigma_1$$

i.e., half those along generatrices.

This is the reason why reservoirs fail along generatrices (Fig. 178*b*).

Longitudinal ribs (Fig. 178*c*) contribute very little to the strength and rigidity of reservoirs, only to the extent of their resistance to bending in the longitudinal plane.

More advantageous are circular ribs (Fig. 178*d*) which work in tension.

The shape of the cylinder ends is of great importance. Flat ends (Fig. 179*a-c*) are unacceptable for high internal pressures. Concave ends are stronger and more rigid (Fig. 179*d-f*). However, the pressure-oriented deformations will cause additional tensile stresses in the cylinder course. Moreover, the concave end appreciably diminishes the reservoir usable capacity.

Conversely, convex ends (Fig. 179*g*, *h*) and similar to them conical ends (Fig. 179*i-k*) restrain radial deformations in the course.

Shields and screens. The rigidity of covers, shields, screens, panels, dashes, etc., is enhanced by flanges (Fig. 180*a-g*), embossing reliefs (Fig. 180*h-k*) and by making them convex (Fig. 180*l*, *m*).

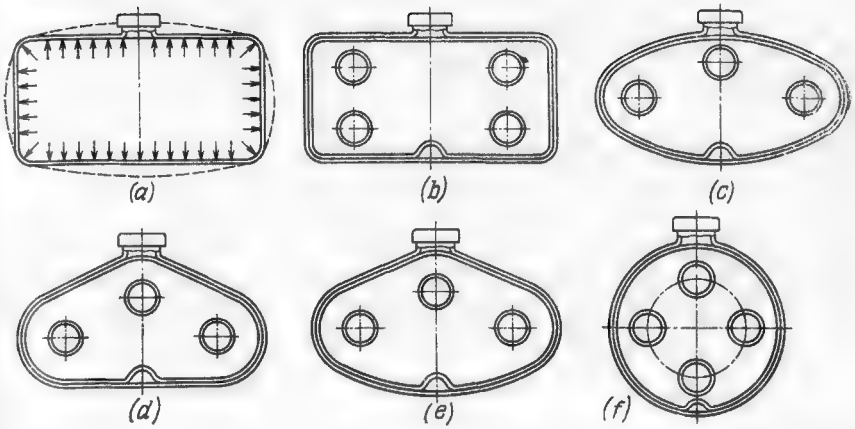


Fig. 177. Enhancing the rigidity of reservoirs subjected to internal pressure

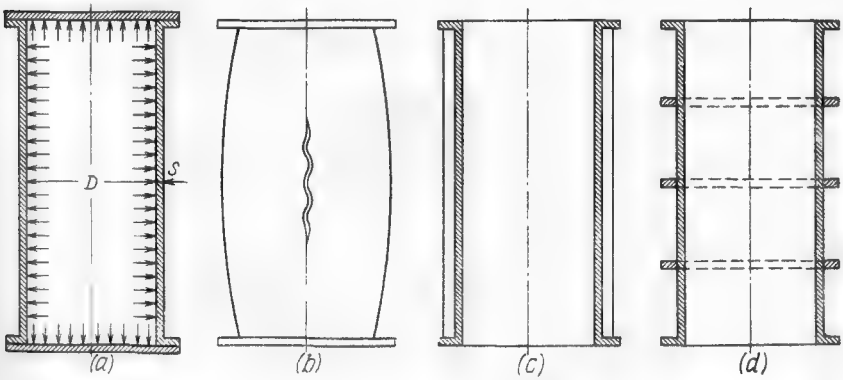


Fig. 178. Arrangement of ribs on the walls of reservoirs subjected to internal pressure

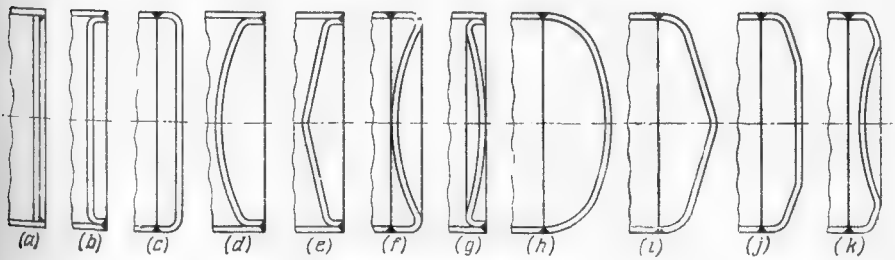


Fig. 179. End shapes

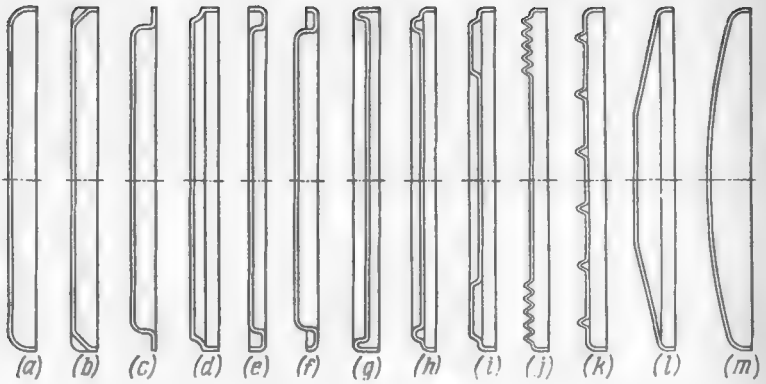


Fig. 180. Designs of panels and screens

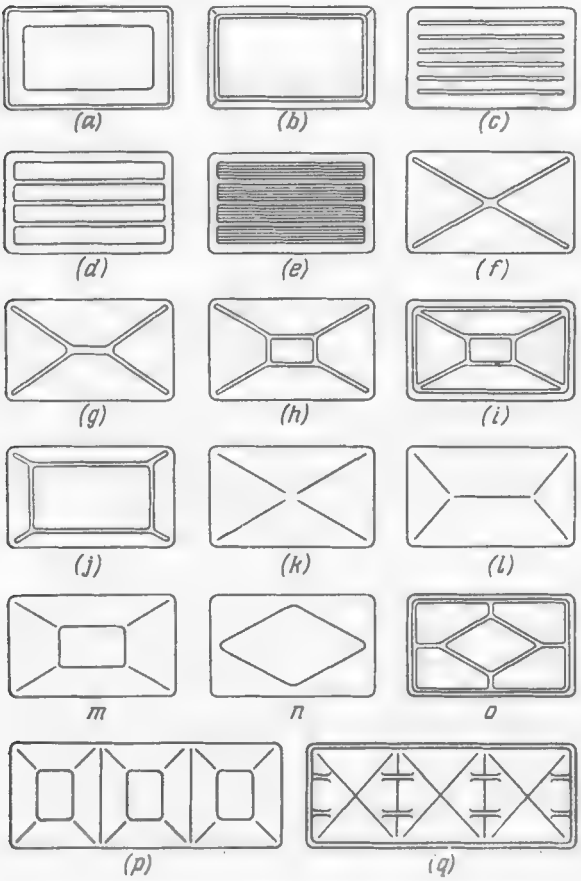


Fig. 181. Plan views of shields and screens

Illustrated in Fig. 181 are typical shapes of covers (in plan) provided with rectangular (Fig. 181*a-e*) and diagonal (Fig. 181*f-j*) patterns of relief. Also shown are covers with diamond convex patterns (Fig. 181*k-o*).

The pattern of relief is often dictated by aesthetic requirements, particularly when faces are in view. Pleasant to look at and sufficiently rigid are rustic forms of surfaces.

Very large shields are usually made of several compartments each of which is strengthened by one of the described methods (Fig. 181*p*). For strengthening longitudinal rigidity compartments are interconnected by a frame or by longitudinal ribs (Fig. 181*q*).

Chapter 5

Cyclic strength

Any part subjected to a sustained repeated-alternating load will fail under stresses which are well below the ultimate strength the material displays when exposed to static load. This feature can hardly be overestimated when dealing with modern multirotary machines, whose parts run under continuous cyclic loads with a total number of cycles throughout the entire period of machine service reaching many millions.

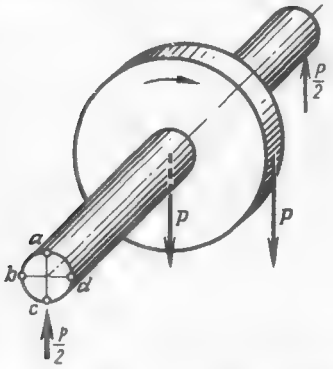


Fig. 182. Diagram of the development of cyclic loads in a gear shaft

According to statistical data at least 80% of failures and accidents encountered when using modern machines are from fatigue phenomena. That is why the problem of fatigue strength is the key enhancing dependability and service life of machines.

Cyclic loads manifest themselves most notably in machines and mechanisms whose parts perform reciprocatory movements (piston-operated machines, cam-actuated mechanisms, etc.). However, cyclic loads are inevitable even in machines with smooth running parts (e.g., rotary machines of turbine type) due to disbalance of rotors, radial and end play of rotors, etc.

With rare exceptions all modern machines incorporate gear drives whose teeth are always subjected to cyclic loading. Also subject to cyclic loads are shafts working under constant directional loads (such are, for example, the shafts of gear-, belt- and chain-gearings).

Let us consider, for instance, a simply-supported gear shaft (Fig. 182) subjected to a driving force P causing flexure in a constant plane. As the shaft makes one revolution, the four points a, b, c and d on the shaft successively move through the plane. With each revolution the cycle repeats. Thus, despite the constant force typical cyclic loads occur.

In short, static loads are exceptions in modern machines. Generally loads change cyclically with a large or small frequency and amplitude.

The number of load cycles which the material can sustain without failure depends on the maximum stress and also on the interval between the extreme values of cyclic stresses. With a stress decrease

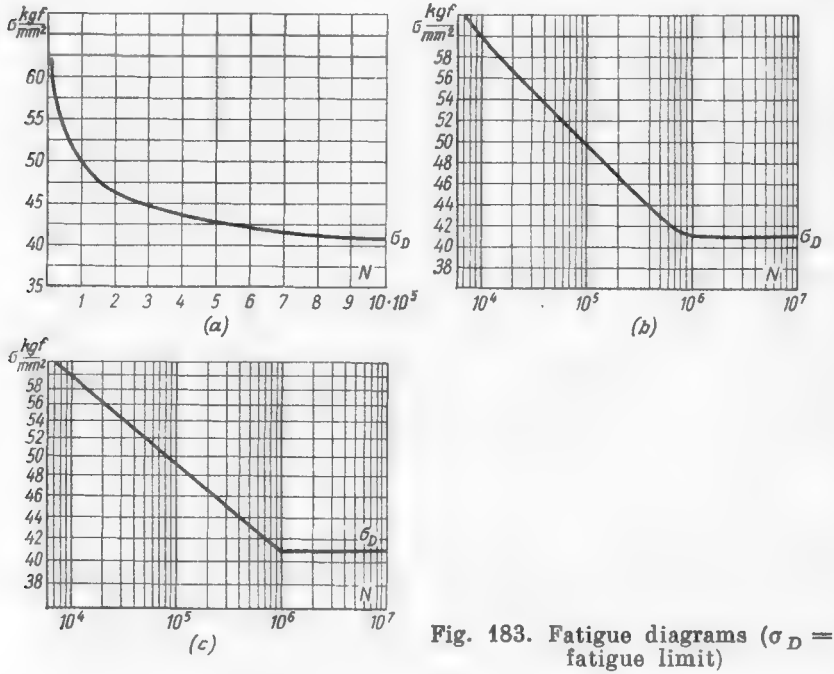


Fig. 183. Fatigue diagrams (σ_D = fatigue limit)

the number of cycles necessary to cause failure will increase and at some sufficiently small stress value the material acquires the ability to withstand an infinite number of cycles without failure. This stress value is called the fatigue limit, assumed as the basis for strength calculations of parts subjected to cyclic loading.

The fatigue limit is ascertained by plotting fatigue curves. The number of cycles N is plotted on the abscissa, and on the ordinate found from testing standard specimens the maximum stresses σ of the cycle which cause failure at a given number of cycles. In the domain of small values the rupture stress approximates the static strength. With the increase in the number of cycles this value decreases. At a certain number of cycles the rupture stress becomes constant.

The ordinate of the horizontal part of the curve is the fatigue limit.

Fatigue graphs are plotted in σ, N coordinates (Fig. 183*a*), in σ, $\log N$ semilog coordinates (Fig. 183*b*) and in $\log \sigma$, $\log N$ log coordinates (Fig. 183*c*). The first method is now almost out of use since it does not allow the shape of the fatigue curve to be ascertained in the region of small or large numbers of cycles. Most often used is the semilog coordinate method.

For most structural steels the fatigue limit becomes evident at 1-10 megacycles. These figures are taken as the basis for determining the fatigue limit of steels (the so-called basic number of cycles). For non-ferrous alloys (e.g., aluminium alloys) the number of load alternation cycles required to determine the fatigue limit is substantially higher (50-100 megacycles). Even after this number of cycles a further drop of fatigue limit is often observed, which shows that in reference to some metals the fatigue limit, as formulated above, does not exist. In this case the nominal fatigue limit is determined as the stress not causing specimen failure within a certain number of cycles (usually 50 megacycles).

There are also no clearly expressed fatigue limits for contact stresses, for cyclic loads in conditions of increased temperatures, and for components working in a corrosive medium. Under such conditions the rupture stress value will continuously fall with the increase in number of cycles. The absence of a clearly defined fatigue limit for large-sized parts is also noticed.

(*a*) *Stress Cycles*

Four basic types of stress cycles are distinguished:

symmetrical alternating — the highest and the lowest stresses are opposite in sign and identical in value (Fig. 184*a*);

asymmetrical alternating — the highest and the lowest stresses are opposite in sign and differ in value (Fig. 184*b*);

pulsating — the highest and the lowest stresses have the same sign but differ in value (Fig. 184*c*, *d*);

complex — various combinations of the above-described cycles (Fig. 184*e-g*).

The cycles have the following fundamental characteristics:

σ_{max} — the greatest algebraic stress value in a cycle (tensile stresses are considered positive, compressive stresses — negative);

σ_{min} — the smallest algebraic stress value in a cycle;

$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$ — mean stress of a cycle;

$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$ — amplitude of stresses in a cycle (the value $2\sigma_a$ is termed cycle stress fluctuation swing);

$r = \frac{\sigma_{min}}{\sigma_{max}}$ — cycle asymmetry ratio (cycle stresses are taken with algebraic sign).

For a symmetric cycle $r = -1$; for asymmetric $-1 < r < 0$; for pulsating (non-symmetric, of fixed sign) $1 > r > 0$; for pulsating in which the maximum or minimum value equals zero, $r = 0$.

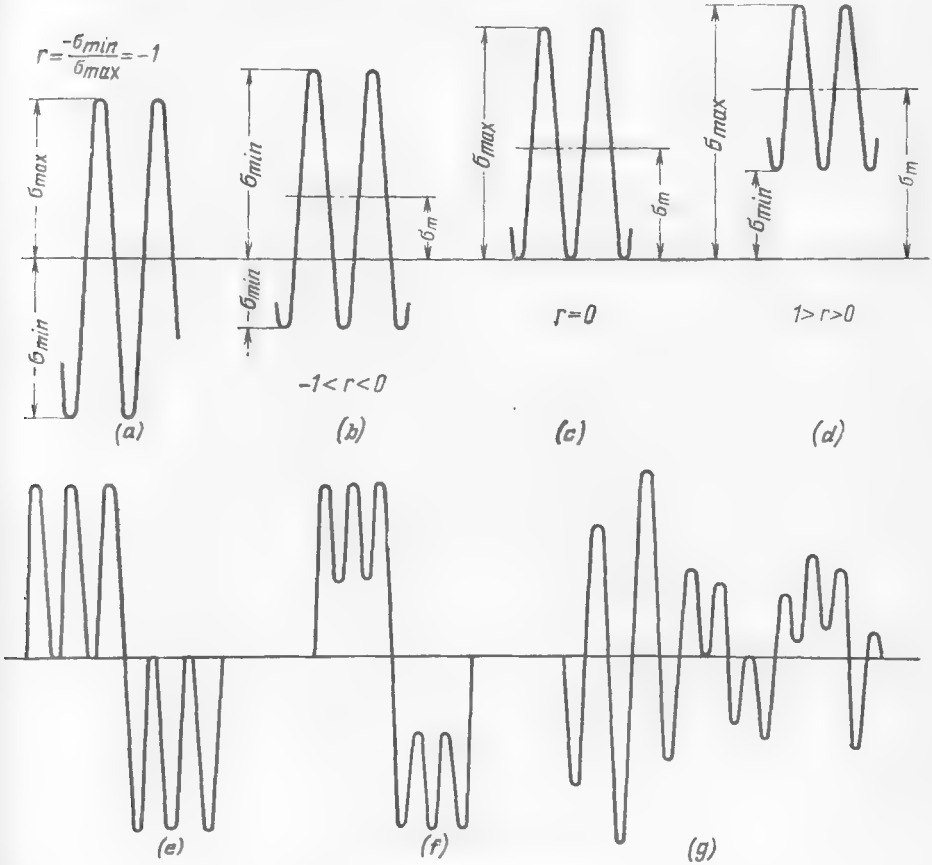


Fig. 184. Stress cycles

The fatigue limit for symmetric cycles is indicated by: in flexure (bending) σ_{-1}; tension-compression σ_{-1p}; torsion τ_{-1}; for pulsating cycles respectively σ_0, σ_{0p} and τ_0.

The most used method for determining the fatigue under symmetric cyclic flexure is that of Weller. A cantilevered or simply-supported sample revolving at a constant speed about its own axis is loaded by a constant directional force. For each revolution all points on the sample surface in the critical section pass once through the zone of maximum tensile stress and once through the zone of maximum compressive stress, making complete cycle of sym-

metrical alternating flexure. The cycle frequency is equal to the number of revolutions the specimen makes in unit time; the number of revolutions prior to failure is the number of cycles causing failure.

Such a kind of flexural loading (termed circular bending) is natural for many engineering parts (e.g., gear shafts, belt and chain drives, etc.).

It should be emphasized that under such type of loading the material behaviour will radically differ from other types of repetitive bending (loading a stationary part with a symmetric cyclically alterating load invariant in direction). In the latter case the fatigue loading will be exerted to only two diametrically opposite zones in the plane of the acting bending moment, whereas with circular bending all peripheral zones of the section will be successively loaded. This cannot but affect the sample life. The compressive-tensile stresses, while travelling over the sample periphery in a crescent-encompassing like motion, will inevitably involve the entire periphery of the sample; under these circumstances each point on the sample surface, taken across the critical section, will be subjected (apart from maximum stresses emerging when the point passes through the bending moment plane) additionally to the action of stresses, which are successively arriving and leaving during sample rotation.

Moreover, the stresses occurring under circular bending, while overlapping fully the periphery of the sample section, will "feel" the weakest points, which may give origin to fatigue cracks; in a stationary sample the weak points are not necessarily found in the bending moment plane.

On the other hand, with circular bending, those portions of material which are leaving the loaded zones are subjected to periodic thermal relief. Under plane bending the loaded portions are constantly under load.

(*b*) *Limited Durability*

The left-hand descending branch of fatigue curves corresponds to the area of limited durability. It enables the durability to be determined (in cycles), which will be typical of components loaded with stresses surpassing the fatigue level, or with such stresses which will be ultimate for a given durability.

Fatigue curves in the zone of limited durability can be, within certain limits, expressed by the equation

$$\sigma^m N = C \qquad (5\ 1)$$

or

$$\sigma = \frac{C}{N^{\frac{1}{m}}}$$

where N = number of cycles
m = exponent
C = constants

The values of m and C can be derived from σ_D and N_0 (i.e., fatigue limit and number of cycles corresponding to such limit), as well as from the original values of σ_1 and N_1 (i.e., original stress, close to the yield limit σ_y and preset number of cycles, Fig. 185).

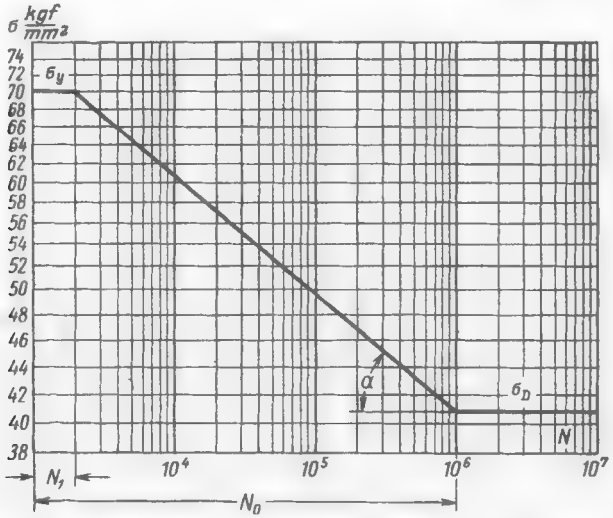


Fig. 185. Determining exponent m of a fatigue curve

For these two points

$$\sigma_1^m N_1 = C \tag{5.2}$$

$$\sigma_D^m N_0 = C \tag{5.3}$$

Having equated Eq. (5.2) and Eq. (5.3) we obtain

$$\sigma_1^m N_1 = \sigma_D^m N_0$$

or

$$\left(\frac{\sigma_1}{\sigma_D}\right)^m = \frac{N_0}{N_1}$$

Having taken the logarithm, we find that

$$m = \frac{\log \frac{N_0}{N_1}}{\log \frac{\sigma_1}{\sigma_D}} \tag{5.4}$$

Substitute the value of m in Eq. (5.3), then

$$C = N_0 \sigma_D^{\frac{\log \frac{N_0}{N_1}}{\log \frac{\sigma_1}{\sigma_D}}}$$

In log coordinates factor m is equal to cotangent of angle α, the inclination of fatigue curve descending branch relative to the abscissa

$$m = \frac{\log \frac{N_0}{N_1}}{\log \frac{\sigma_1}{\sigma_D}} = \frac{\log N_0 - \log N_1}{\log \sigma_1 - \log \sigma_D} = \cot \alpha$$

Generally the log scale for stresses is larger than the scale for the number of cycles. In this case

$$m = \frac{\log N_0 - \log N_1}{a (\log \sigma_1 - \log \sigma_D)}$$

where a is the coefficient accounting for differences in scales.

The values of m depend on material properties and shapes of components. On the average, for plain components m = 8-15, while for components with stress concentrators m = 3-8. The m-parameter may, to a certain degree, be regarded as a resistance criterion of material fatigue. The smaller the m value (i.e., the steeper the fatigue curve slope), the shorter the service life of parts at stresses exceeding the fatigue limit and, as a general rule, the lower the fatigue limit.

Generally, components designed for a limited service period are those made of materials lacking a definite fatigue limit or having a steeply dropping fatigue curve (concentration-sensitive materials), as well as the parts which, because of size or weight considerations, cannot be given dimensions determined by the fatigue limit. Considered in the same way are machines and mechanisms operating under low-cycle conditions and mechanisms in which operative periods alternate with prolonged downtimes or operation under low loads (e.g., hoisting machines of intermittent action, etc.), i.e., mechanisms in which the total number of cycles throughout the entire service period is less than the number of cycles corresponding to the fatigue limit.

Parts subjected to high-frequency loads of continuous action are considered against the fatigue strength limit with the necessary safety factor. The service life of such parts is sharply reduced if the fatigue limit is exceeded.

Assume that a mechanism works at 2000 rpm, i.e., its parts are subjected to 2000 cycles per minute. Its service life limit is determined according to Eq. (5.1) from the relation

$$\frac{N}{N_0} = \left(\frac{\sigma_D}{\sigma}\right)^m \qquad (5.5)$$

Let $N_0 = 10^6$ cycles, $m = 5$. From Eq. (5.5) we find that at stresses reaching 1.5, 1.2 and 1.1 times that of the fatigue strength limit $\left(\frac{\sigma_D}{\sigma} = 0.666;\ 0.833;\ 0.91\right)$, the service period will be equal to 1 h, 3 h 20 min and 5 h, respectively. At stresses equal to the fatigue limit the service life becomes infinite. Thus, even the most insignificant increase of designed stresses over the fatigue strength limit severely lowers service life and does not save weight or size.

(*c*) *Fatigue Limits*

Fatigue strength limit cannot be regarded as a constant characteristic inherent in a given material. It is actually subject to much greater fluctuations as compared with mechanical properties of material under static loading. The magnitude of this limit is dependent on the conditions of loading, type of cycle (particularly on the degree of its asymmetry), test methods, shape and absolute dimensions of a part, manufacturing conditions, state of a surface, etc.

Thus, it is more correct to say that when testing standard samples for fatigue it is not the fatigue strength of a material but the fatigue limit of a sample made of this material which is being tested. When passing from a test sample to a real part it is necessary to introduce corrections which consider the shape and size of parts, their surface condition, etc. Therefore, we may speak about fatigue strength of parts.

In this connotation the fatigue strength limit implies a rather different notion in contrast to its original meaning of a material characteristic, although the fatigue limit determined by testing standard samples is still quoted as one of the basic strength properties of materials.

Now we speak also of fatigue strength of units (threaded, press-fitted and other assemblies). Thus, the fatigue strength notion includes not only the factors of material properties and geometry of parts, but also the interaction factors of adjacent components.

Full-scale tests become more popular in which service periods and fatigue strength limits of parts and assemblies are determined.

Flexure fatigue limits have their minimum value during a symmetrical alternating cycle, increase with the growth of cycle asymmetry, rise in the range of pulsating loads and approach to the values of material static strength with the decrease of pulse amplitudes.

Fatigue strength to tension-compression is approximately 1.1-1.5 times more, and in torsion 1.5-2 times less than for the case of a symmetrical alternating flexure.

No definite relationship exists between the characteristics of fatigue and static strength. The most stable relationships exist

between σ_{-1} (fatigue limit in flexure with a symmetric cycle) and σ_b (ultimate strength) and also $\sigma_{0.2}$ (yield limit under static tension).

Experiments give the following correlations:

for steel

$$\sigma_{-1} = (0.2 - 0.3)\,\sigma_b \left(1 + \frac{\sigma_{0.2}}{\sigma_b}\right)$$

for steel castings, high-strength cast iron and copper alloys.

$$\sigma_{-1} = (0.3 - 0.4)\,\sigma_b$$

for aluminium and magnesium alloys

$$\sigma_{-1} = (0.25 - 0.5)\,\sigma_b$$

for grey cast iron

$$\sigma_{-1} = (0.3 - 0.6)\,\sigma_b$$

Schimek obtained the following graphical relationships between fatigue strength and tensile strength σ_b (Fig. 186) found by analyzing the results of fatigue tests carried out on refined structural steels:

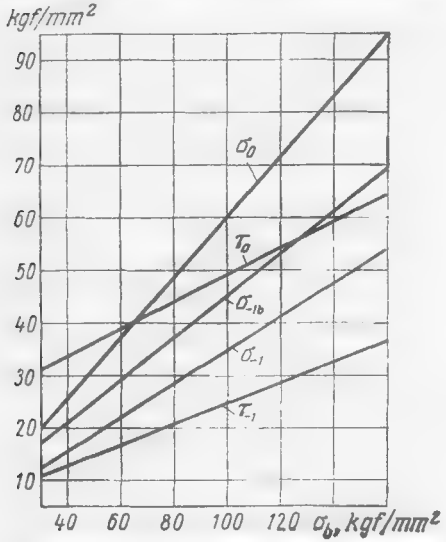


Fig. 186. Fatigue limits for various cycles of loading as a function of tensile strength σ_b (after Schimek)

for tension-compression under a symmetrical cycle

$$\sigma_{-1} = 0.33\sigma_b + 1.25$$

for tension-compression under a pulsating cycle

$$\sigma_0 = 0.58\sigma_b + 2.3$$

for flexure under a symmetric cycle

$$\sigma_{-1b} = 0.4\sigma_b + 5.7$$

for torsion under a symmetrical cycle

$$\tau_{-1} = 0.2\sigma_b + 4.8$$

for torsion under a pulsating cycle

$$\tau_0 = 0.25\sigma_b + 24.2$$

Under a symmetric cycle the fatigue limits are related together by the following tentative relationships

$$\sigma_{-1} = (1 - 1.5)\,\sigma_{-1p}$$

$$\tau_{-1} = (0.5 - 0.7)\,\sigma_{-1}$$

Under a pulsating and alternating symmetrical cycle the fatigue limits are connected together by the following approximate relationships:

in flexure

$$\sigma_0 = (1.4 - 1.6)\,\sigma_{-1}$$

in tension

$$\sigma_{0p} = (1.5 - 1.8)\,\sigma_{-1p}$$

under torsion

$$\tau_0 = (1.4 - 2)\,\tau_{-1}$$

The fatigue limits for asymmetric cycles can be approximately determined by an empirical relationship between maximum stress σ_{max} of cycle, the mean cycle stress σ_m and ultimate cycle amplitude σ_a.

One of such relationships

$$\sigma_{max} = \sigma_{-1p}\left[1 - \left(\frac{\sigma_m}{\sigma_b}\right)^2\right] + \sigma_m$$

and

$$\sigma_a = \sigma_{-1p}\left[1 - \left(\frac{\sigma_m}{\sigma_b}\right)^2\right]$$

where σ_b is the static ultimate tensile strength.

These correlations give some idea about general relationships. For calculations it is necessary to refer to technical literature on fatigue strength.

(*d*) *Generalized Fatigue Diagrams*

The interrelationships between the fatigue strength, mean stress of cycle and coefficient of cyclic asymmetry are shown in the form of generalized diagrams. The most used are Smith's diagrams (Fig. 187). The line of the mean stress cycle $\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$, which is at the same time the zero line of amplitudes, is drawn at 45° to the abscissa; the stress scale is plotted on the ordinate axis. Plotted on the zero line are the stress amplitudes found from experiment and safe for each given value of σ_m and characterized by the stress values σ_{max} and σ_{min}. The *ABC* envelope of the σ_{max} points represents the fatigue limit in tension and the *DEF* envelope of the $(-\sigma_{max})$ points, the fatigue limit in compression.

The higher limit for σ_{max} is assumed to be the tensile yield limit $\sigma_{y\ tens}$ (line *BC*) and for $(-\sigma_{max})$, the compressive yield limit $\sigma_{y\ compr}$ (line *DE*).

The coefficient of the asymmetry cycle $r = \frac{\sigma_{min}}{\sigma_{max}}$ for arbitrary point *a* is determined as a relation of lengths $\frac{bc}{ab}$ (the first correspond-

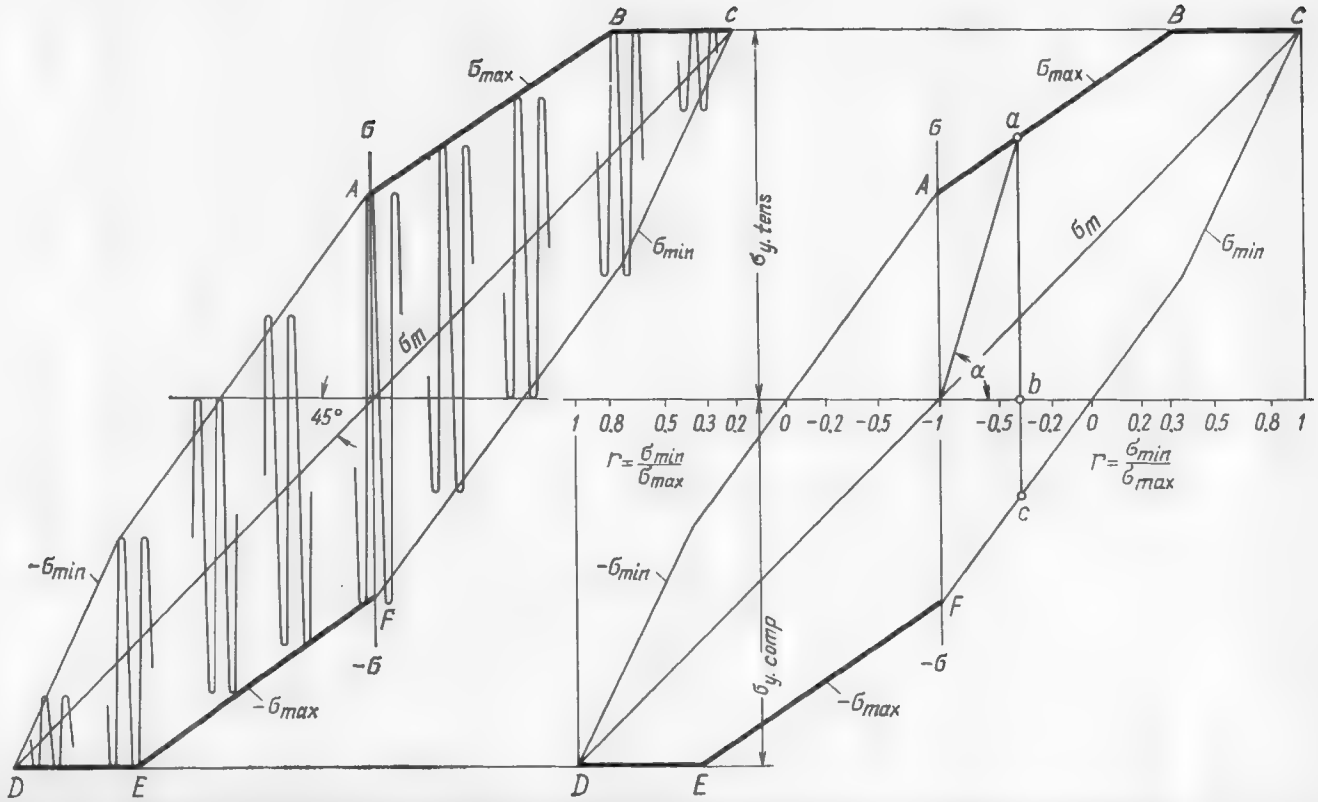


Fig. 187. Smith's diagram (the material has the same tensile and compressive resistances)

ing to σ_{min} and the second, to σ_{max}). Each length is taken with its own sign.

The magnitude of r may also be determined from

$$r = \frac{2}{\tan \alpha} - 1$$

where α is the incidence angle of a beam connecting point a with the origin of the coordinate axes.

By arranging the Smith's diagrams for definite materials and load forms it is possible to perform fatigue calculations for any value of the asymmetry cycle coefficient.

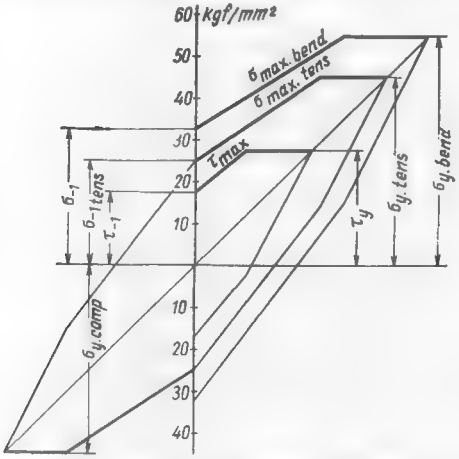


Fig. 188. Smith's diagram plotted for circular bending, cyclic tension-compression and cyclic torsion

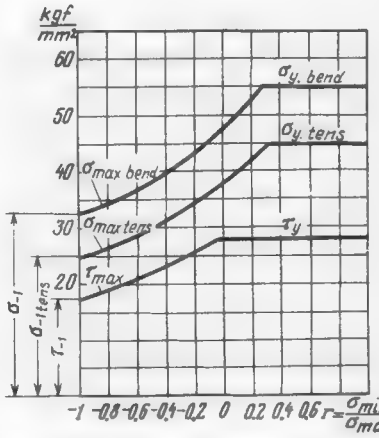


Fig. 189. Fatigue limits in circular bending, cyclic tension-compression and cyclic torsion as a function of cycle asymmetry coefficient r

Shown schematically in Fig. 188 is a Smith's diagram for a structural steel subjected to three kinds of loads: circular bending, cyclic tension-compression and cyclic torsion. Diagrams for bending and torsion are plotted only on one side of the ordinate axis as they encompass in that region all possible kinds of stresses.

In practice it is more convenient to employ diagrams depicting fatigue limits for different kinds of loading as a function of the cycle asymmetry coefficient r (Fig. 189) since these diagrams contain, in concise form, the same data as Smith's diagrams.

(e) *Damageability Curves*

The value of fatigue strength is affected by overloads which the part is subjected to before loading. One method accounting for such overloads, suggested by French, consists in plotting damageability

(French's) curves. The method consists in preliminarily loading the samples with stresses exceeding the fatigue limit at different number of cycles and consequently testing the samples at stresses corresponding to the fatigue limit level.

Let samples be subjected to a test stress which is 1.5 times the fatigue limit at 10^4; $5 \cdot 10^4$; 10^5; $5 \cdot 10^5$, etc. cycles. In the course of the following fatigue test some samples when subjected to an overstress duration, say, higher than 10^5 cycles,fail; while other samples subjected to overstressing for a smaller number of cycles remain sound. This means that at a number of cycles greater than 10^5 some

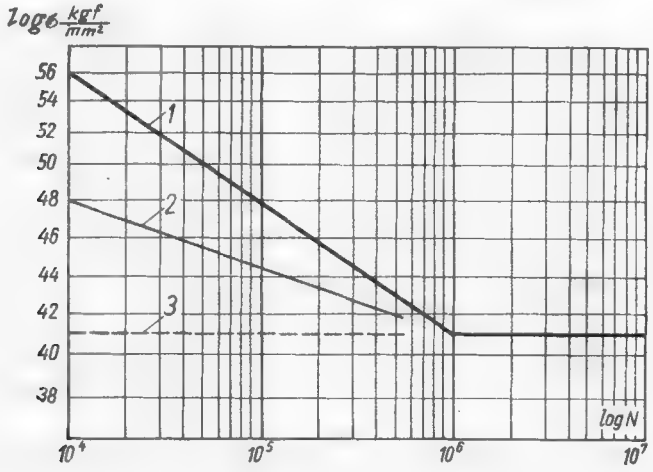


Fig. 190*a*. Damage curves

irreversible damages occur in the metal, which make the part incapable of further cyclic loading, even if the stresses are maintained at the fatigue strength limit. On the other hand, loading duration less than 10^5 cycles is safe. The point corresponding to the stress equal to 1.5 times the fatigue limit and duration of 10^5 cycles is plotted onto the fatigue diagram. The locus of such points taken for different overstress levels and corresponding safe cyclic durations clearly outlines the zone of safe overloads on the diagram.

A typical damageability curve (in $\log \sigma$ *vs.* $\log N$ coordinates) is shown in Fig. 190*a*. Overloads positioned below damageability curve *2* are safe, while those between curves *1* and *2* are inadmissible.

The closer curve *2* to curve *1*, the better the ability of material to withstand overloads. For some high-strength materials subjected to efficient heat-treatment, the second curve *2* of French practically coincides with the slopes of the first curve *1* of Weller. In some other

materials (e.g., annealed carbon steels) the third curve *3* of French is the continuation of the horizontal portion of Weller's first curve *1*. This means that such materials are quite incapable of sustaining

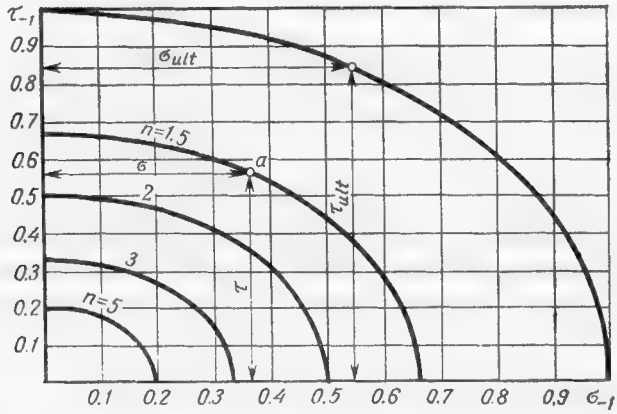


Fig. 190*b*. Relationship between admissible shear stress τ and tension stress σ for biaxial state of stress (symmetric bending and torsion)

overloads. For this reason, parts made from these materials must be calculated by the fatigue limit, even in the region of limited durability.

(*f*) *Fatigue Strength in the Case of Complex States of Stress*

The problem of fatigue strength under complex states of stress is still not fully studied. Investigated better than others is the biaxial state of stress in which simultaneously act symmetrically varying cyclic stresses, both normal and tangential (cyclic tension-compression and torsion, cyclic flexure and torsion). Found experimentally for the case the ultimate values of normal stresses σ_{ult} and ultimate values of shear stresses τ_{ult} can be formulated through an ellipse-type relation:

$$\left(\frac{\sigma_{ult}}{\sigma_{-1}}\right)^2+\left(\frac{\tau_{ult}}{\tau_{-1}}\right)^2=1$$

where σ_{-1} and τ_{-1} are fatigue limits under pure tension-compression and pure symmetrical torsion, respectively.

At a given value of σ_{ult} the admissible magnitude of concurrently acting shear stress

$$\tau_{ult}=\tau_{-1}\sqrt{1-\left(\frac{\sigma_{ult}}{\sigma_{-1}}\right)^2}$$

and, conversely, at a given value of τ_{ult} the admissible magnitude of simultaneously acting tensile-compressive stress

$$\sigma_{ult} = \sigma_{-1} \sqrt{1 - \left(\frac{\tau_{ult}}{\tau_{-1}}\right)^2}$$

This relation is shown graphically in Fig. 190*b* (heavy line). Any combination of stresses positioned between the limiting curve τ_{ult}—σ_{ult} and axes of coordinates (e.g., point *a*) is safe. The margins of safety for any combination can be determined by plotting a network of curves of equal safety with the magnitudes of τ_{-1} and σ_{-1} decreasing proportionally to the safety coefficient n (thin curves).

Diagrams like those presented in Fig. 190*b* have been plotted for symmetrical alternating cycles with cophasally varying values of τ and σ. The regularities typical of these diagrams apply also to asymmetrical cycles, as well as to the cases of acophasal variations of τ and σ.

The fatigue strength behaviour under non-stationary conditions of τ and σ variations, as well as in triaxial stressed conditions has so far been studied insufficiently.

(*g*) *Effect of Load Character on the Fatigue Limit*

The effect of cycle frequency and speed of stress changes within a cycle on the fatigue limit has not been completely studied.

It has been proved that increasing the number of cycles per unit time increases fatigue strength, particularly noticeable at frequencies greater than 1000 cycles per minute. For some materials the following relation is established:

$$N = A\mu^{\frac{1}{2}}$$

where N = number of cycles prior to failure
μ = cyclic frequency
A = constant

Improvement in the fatigue limit with increasing cyclic frequency may be attributed to the fact that plastic strains occur at a low rate (hundreds of times less than the speed of elastic strains equal, as is known, to the sonic speed in a given medium). Increased frequency of cycles suppresses plastic strains in microvolumes of metal preceding the appearance of fatigue cracks.

The main divisions in theory of fatigue include: fatigue under cyclic impact load (impact fatigue), fatigue under cyclic contact load (contact fatigue), fatigue under increasing and periodically varying temperatures (thermal fatigue). The mechanism of fatigue strength under the above-described conditions is still not fully investigated.

(*h*) *Nature of Fatigue Failure*

Fatigue failure is the result of repetitive and quickly alternating elastic and elastic-plastic deformations distributed, due to non-homogeneity of the material, non-uniformly through the volume of the part. Initial failures arise in the microvolumes unfavourably orientated relative to the acting loads, prestressed by residual stresses and weakened by local defects. Such local damages, as they gradually accumulate and pile up, may initiate the process of component general failure.

In the fatigue failure processes a great role is played by the sources of liberated heat in the microvolumes undergoing deformations. As a result of increased temperature the strength of material in the microvolumes lowers. This facilitates the inception of additional plastic shears which, in their turn, cause further rises in temperature.

The greater the heat liberated in the microvolumes, the larger the stress amplitude and the less the cycle asymmetry factor. On the other hand, the magnitude of local rise of temperature is dependent on the properties of material and on its structural constituents. The greater the increase in temperature in a microvolume, the less the heat-conductivity and heat-capacity of the material and the higher its cyclic toughness which (at the stage of elastic deformations) determines the amount of irreversible conversion of vibrational energy into thermal energy.

This explains why the fatigue strength has its minimum value in the case of symmetrical cyclic stresses causing the greatest oppositely directed shears. This also explains why high but short-time overloads do not lower fatigue strength: the heat evolved in the overstressed microvolumes quickly dissipates into the surrounding masses of material, thus enabling the strength of overstressed volumes to be recovered.

The initiation process of a fatigue crack has several stages. At the initial stages of loading cracks originate at the boundaries of crystallites (grains) as a result of displacements (shears) of packs of crystalline planes oriented in parallel to the action of maximum tangential stresses, i.e., directed approximately at 45° to the tensile stresses (octahedral shearing stresses). Depending on the orientation of a grain the shears can occur in one plane, or simultaneously in two (Fig. 191*IIIa*, *b*) or three (Fig. 191*IIIc*) planes.

At a certain loading stage the mass of metal presents a mosaic of grains subjected to plastic strains (Fig. 192), and grains free of plastic strains thanks to more favourable orientation of crystalline planes relative to the action of tangential stresses.

The inception of initial cracks within the boundaries of a grain is essentially the result of a directed propagation (diffusion) of dislocations (such as vacancies) toward the grain boundaries. The

rate of diffusion is proportional to the magnitude of stresses and temperature. The diffusion process is intensified by microheating of metal.

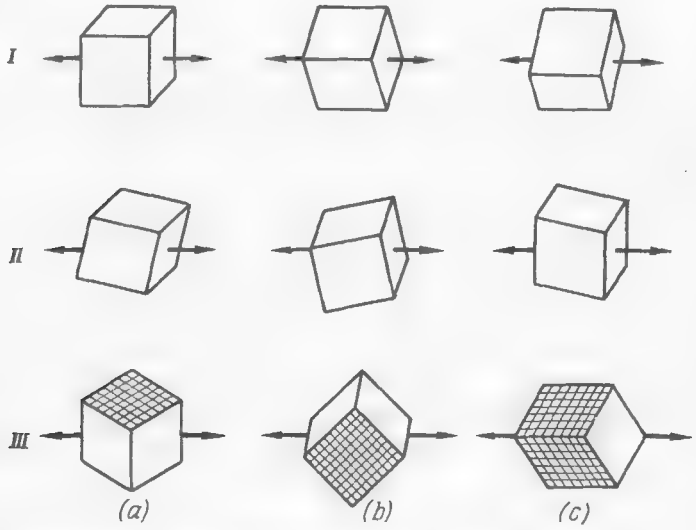


Fig. 191. Orientation of crystallites relative to active forces
I-II — favourable; *III* — unfavourable

The accumulation of vacancies leads to loosening of the structure, formation of submicropores and, finally, the appearance of initial cracks.

At the initial stages the process is reversible. As soon as stresses discontinue (i.e., during intervals of repose), vacancies migrate

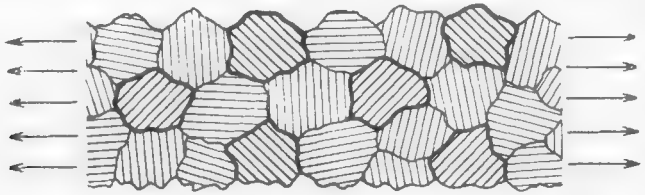


Fig. 192. State of stress in a surface layer subjected to a tensile load (heavy lines discriminate the grains with crystal planes parallel to tangential stresses

in the reverse direction. As a result, cluster vacancies are gradually reabsorbed and distributed uniformly throughout the grain microvolumes. Thereby, the material returns to its original state. The process can be stimulated by raising the temperature. Experience proves that initial disruptions can be cured by short-time heating.

If the stresses continue to act, the disruption accumulation process develops. Gradually propagating, initial cracks emerge to the grain surface. Here their development stops mainly due to obstacles being created by other crystalline orientation of adjacent grains. This disorientation of crystalline planes leads to plastic shear displacement.

Grain interlayers due to their inherent impurities possessing a strongly distorted crystalline lattice, sometimes different in type from the grain crystalline lattice, serve as another obstacle.

Thus, self-formed intergrain barrier is produced which effectively brakes crack propagation. To overcome this barrier, stress is required significantly in excess of the stress causing intergrain shear. The magnitude of the penetrating (break-through) stress depends on the interlayer strength and on the degree of disorientation of the grain crystalline planes separated by the interlayer. Obviously, interlayers between grains with identically oriented (directed) crystalline planes are the easiest to overcome. However, these cases are statistically rare.

The mean value of stress necessary to overcome intergrain barriers determines the fatigue strength of the material. The fatigue limit can be regarded as the average stress level at which the crack nuclei still remain within grain boundaries and are partially or totally restored during rest intervals.

The resistance of material to intergrain shear depends on its physical and mechanical properties and the fine crystalline structure of a grain. Of great significance is the size of minute (some hundredths of a micron) crystalline clusters (subgrains) comprising the grain. Diminution of crystalline clusters and their still greater disorientation, as well as distortions of the atomic-crystalline lattice caused by impurities, strain hardening, precipitation of secondary phases and inception of non-equilibrium (hardenable) structures — all these factors enhance the resistance to intragrain shear and improve the fatigue strength of material. Actually, it is this that the strengthening effect of alloying, heat treatment and plastic deformation is aimed.

Should the level of stresses related to the entire thickness of the material, be below the fatigue limit, then the initial cracks may practically remain for an unlimited time within individual grains without causing significant loss in part strength. If the stresses throughout the entire part or in part of its volumes exceed the fatigue limit (e.g., due to local stress concentrations), then the cracks will overcome intercrystallite barriers and extend into the mass of metal.

As soon as it extends beyond the grain boundaries, the crack intermittently widens developing into a macrocrack which changes the direction of its propagation extending now normally to the action of maximum tensile stresses. The development of the crack is now

expedited by the onset of abrupt stress concentrations which occur at the crack leading edge. Local failure entails heating which softens the metal, thus, contributing to the crack propagation.

A macrocrack can extend under the action of stresses well below those necessary to overcome the interbarrier resistance, in addition the stresses necessary for crack propagation lessen as the crack grows.

Having reached the part surface, the crack then starts to penetrate into the depth of material progressing along the material weakest regions.

Simultaneously, a large number of cracks develops. However, at a definite stage, the process is localized: mainly one or a group of local cracks expands, having outstripped the rest in their development by virtue of material defects concentrated at the given place, localized prestressed rupture, or on the strength of unfavourable orientation of crystals relative to the acting stresses.

The adjacent cracks blend together forming a deep branched system. No new plastic shears or cracks come into existence, while those having had time to occur either discontinue or slow down their development as all strains are taken up by the main crack. Propagation of the main crack results in the part failing owing to decrease of its net-section.

Contrary to the first stages in the appearance of intergrain and intragrain cracks developing over a long period final failure begins abruptly and has the form of brittle fracture.

Fatigue fractures generally display two sharply visible zones. The zone of fatigue crack propagation has a dull porcelain-like surface typical of fractures in which transcrystallite failures are predominant (the so-called state-pattern fracture). Crack edges often show smooth, shiny work-hardened regions—the result of impacts, crushing and abrasion of crack walls during the periodic deformations of the material.

The zone of final failure has a crystalline surface typical of brittle fractures in which intercrystallite failures are predominant (e.g., impact fractures and fracture of brittle materials). A streak-like pattern is usually seen in the zone of failure, i.e., a series of parallel lines which are, in fact, traces of intermittent advance cracks in proportion to the cycle accumulation of alternating loads.

As a rule, initial cracks originate, for all forms of loading, in the surface layer whose thickness, on the average, does not exceed three grain diameters (which amounts to 0.05-0.2 mm for steels). Usually cracks occur in grain fragments located on the surface and cross-cut during preceding machining.

Thus, for fatigue strength the surface layer plays the decisive part. It is particularly important because in the majority of loading cases (flexure, torsion, complex stressed states) it is the surface layer that is subjected to the maximum stresses.

A number of reasons explain the particular role of the surface layer.

Firstly, the pure physical factors should be noted. As known from the physical regularities, the packing of atoms in the surface layer is closer than those lying below.

As a result of the interaction with the underlying less densely packed layers, in the surface layer there develop tensile stresses giving rise to loosened spots which are potential sources of the crack formation.

Secondly, the metal particles which emerge to the surface, possessing only one-way metallic bonds with the underlying metal, have a greater activity and easily combine with particles of the surrounding medium. On the bare surface of the metal there are formed stable adsorbed films of vapours, gas, moisture, oxides, etc. which cannot be removed by mechanical or chemical methods. Penetrating deeper into the depth of the material through microcracks, the adsorbed films disturb the solidness of the metal, and weaken the surface layer. Of much significance is the cleaving action of thinnest films of surface-active agents infiltrating into submicroscopic slits on the metal surface. With slits widths of the order of hundredths of a micron the films may build up extremely high pressure (occasionally reaching hundreds and thousands of atmospheres), and contribute to metal failure.

Thirdly, some processing factors must be mentioned. The surface layer inevitably, to a greater or less degree, is impaired by the preceding processing. Machining, even the finest, inevitably causes radical changes in the surface layer. Being actually a combination of plastic deformation and destruction of metal, the machining procedure is accompanied by grain cutting, breaking and tearing of individual grains, appearance of microcracks and onset (in the surface layer and adjacent regions) of high residual rupture stresses, close to the material's yield limit. The heat evolved in the machining process induces partial recrystallization of the surface layer and sometimes is accompanied with phase and structural changes.

The heat treatment process is often followed by surface layer decarburization, decomposition of pearlite and cementite with the formation of unstable ferrite scale.

Fourthly, the metal surface is attacked by all kinds of corrosion encountered in practice and causing deep damages to the surface layer. Usually corrosion spreads along grain interlayers and microcracks.

Any surface working in friction is subjected to one more form of weakening—wear. Wear lowers fatigue strength significantly and is accompanied by changes in microgeometry and disturbances in the surface layer structural pattern.

Thus, concentrated in the surface layer are numerous and diverse

submicro-, micro- and macrodefects, which are caused by mechanical, physical and chemical factors unavoidable due to the technical conditions of producing the surface layer as well as the particular role of the surface layer as the surface separating the metal from the surrounding medium. One may rightfully state that the surface layer of each part is a stress concentrator (or stress raiser), whose effect can be lessened by a number of measures but which can never be eliminated completely.

All those factors, that disturb the continuity and homogeneity of the surface layer causing the appearance of higher tensile stresses, promote the onset and development of initial cracks and sharply lower the fatigue strength of a material. Conversely, compaction of originally loose structure of the surface layer and creating preliminary compressive stresses in it even at small depths (shot-blasting, rolling) materially enhance the resistance of material to cyclic loads.

Strength under cyclic loadings can also be improved by removing the defective surface layer by some techniques which do not introduce further damage (microfinishing, polishing). It has been found that deep repolishing of specimens in the course of fatigue tests sharply improves their service life. This is attributed to the partial removal of the surface layer together with the initial cracks formed in it at the previous testing stage and packing of the surface layer by repolishing and partial curing of the existing microcracks.

Therefore, the improvement of fatigue strength requires first of all work-hardening of the surface layer. This is obtained by chemical heat treatment processes, surface thermo-diffusion alloying, surface strain hardening, etc. Of much importance is the elimination of macro- and micro-defects in the surface layer, particularly defects brought about as a result of machining.

In hollows (e.g., tubular components) being subjected to tensile or complex stresses in which tensile stresses are predominant, the conditions of the inside surfaces are no less important than those of the outside surface. In such components the inside surfaces should be subjected to strain hardening and carefully checked for defects.

Experience proves that fatigue strength (in contrast to static strength) only slightly depends upon grain size, which seems at first glance paradoxical: fine-granular metals with deep strengthening lattice of cleavage surfaces would appear to resist cyclic loads better than metals possessing coarse grains and loose lattice. In actual fact this phenomenon is quite natural.

Fatigue strength is determined by the stress necessary to overcome the initial intercrystallite barriers. As soon as these barriers have been broken through, the propagation of a crack is materially facilitated. Widening, the primary crack propagates in a way typical of macrocracks, easily crossing all the following barriers (at moderate temperatures, usually in a transcrystallite mode).

(*i*) *Stress Concentration*

The fatigue strength of parts is seriously aggravated by the presence of weakened spots, sharp transition areas, entering angles, etc., leading to local stress concentrations. The abrupt changes of stresses occurring at weakened spots may exceed 2-3 and more times those average values which are commonly encountered in the same section of a part (Fig. 193*a*).

Inasmuch as severity of initial fatigue damage is dependent on the diffusion rate of vacancies, while the latter is proportional to

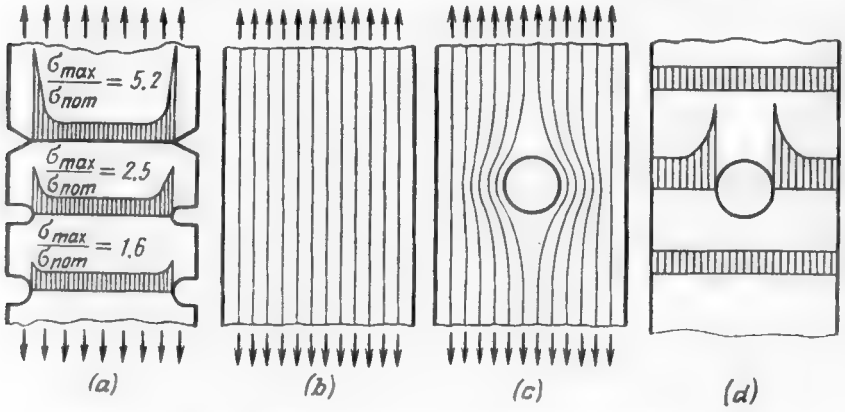


Fig. 193. Force flow in a part undergoing tension

the magnitude of active stresses, it is not surprising that at the spots of stress concentrations the metal becomes loose, this generally preceding the inception of fatigue cracks. Due to this the fatigue damage in the stress concentration zones outstrips damages in the remaining parts of the piece.

The intensity of stress increase depends primarily on the kind and form of weakening. The greater the section gradient at the part of transition and the sharper the transition, the higher the local maximum stress.

The stress concentration phenomenon has been carefully studied—both theoretically and experimentally.

Given below is a simplified description of how stress concentrations occur. The concept is based on the phenomenon of a distorted force flow which occurs in a weakened zone. Though not reflecting all the complexity of the picture, the suggested scheme seems to be rather illustrative and true, thus enabling certain practical conclusions to be drawn.

Let us assume that a beam is subjected to a tensile load P (Fig. 193*b*) and that the latter is uniformly distributed throughout

the entire sectional area. Through the agency of internal bonds the load applied to each sectional point is transmitted to adjacent points.

The paths of load transmission from one point to another along the beam body are called force lines (shown by fine lines in the figure) and the aggregate of these lines is referred to as force flow.

The force lines are continuous and cannot be broken since this would signify local breakage of bonds between adjacent points, i.e., the beginning of material failure. Hence, the number of force lines must be the same at any cross section.

The density of force flux (the number of force lines per cross section unit area) defines the stress value. If the part sectional area reduces (e.g., due to a central bore in the part body—Fig. 193*c*), then in

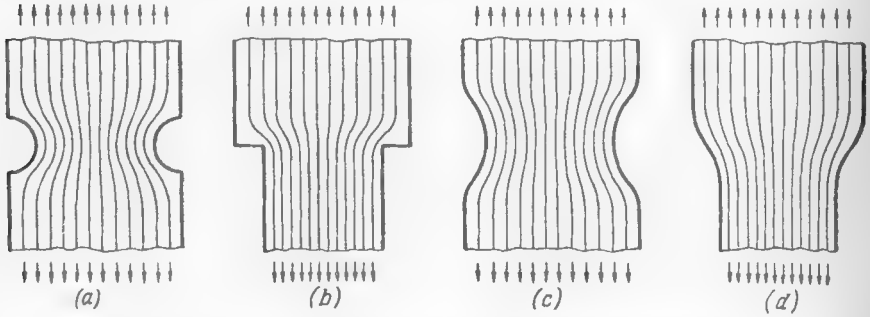


Fig. 194. Force flow in a part with changing sections

constricted sections the force lines get thickened causing a rise in the stresses. This is always accounted for when carrying out strength calculations at the design stage.

At the same time the force lines bend while passing the obstacle. This means that under the influence of tensile loads the material fibres positioned close to the obstacle are subjected not only to tensile stresses but to bending as well.

In the event of an internal hole the fibres being bent by tensile forces will tend to converge to the centre, thus producing transverse material compression at the hole, as if contracting the latter.

These stresses when added to the tensile stresses in a weakened region close to the hole cause a sharp stress increase (Fig. 193*d*).

Analogous picture is seen in the case of recesses positioned along the sides of a part (Fig. 194*a*). Here the force lines are bent (curved) in the recess area. Here the bent fibers tend to spread the sides, causing transverse tension of the material near the recesses as if opening them.

In the event of abrupt stepped-like changes of sections (Fig. 194*b*) the force lines are also bent. At areas above the transition (shoulder) the material undergoes transverse compression, while in areas below

the transition the material is subjected to transverse tension. The fibers under tension tend to open the transition entering angles, thus producing local tensile stresses of higher values. The intensity of transverse tensile-compressive stresses is proportional to the slope angle of the force lines.

Tensile stresses can be considerably reduced by making transitions streamlined (Fig. 194*c*, *d*): this decreases the force line curvature and increases the strength of the weakened portion.

Pictures similar to those illustrated in Figs 193-194 are also observed in cases of flexure and torsion in parts with local weakenings.

From the above it is clear that one active way of lowering stresses in weakened sections is to give the transition portion smooth outlines.

A well-known positive effect is produced by stress deconcentrators, i.e., additional local weak spots made close to the chief source of stress concentration.

In a part subjected to tension and having a stress raiser in the form of a central hole (Fig. 195*a*), stress deconcentrators can be made in the form of additional small-diameter holes located in line with the main hole along the force flux direction (Fig. 195*b*). Passing the obstacle the force lines curve less than in the case of one central hole. Consequently, the stress concentration lowers.

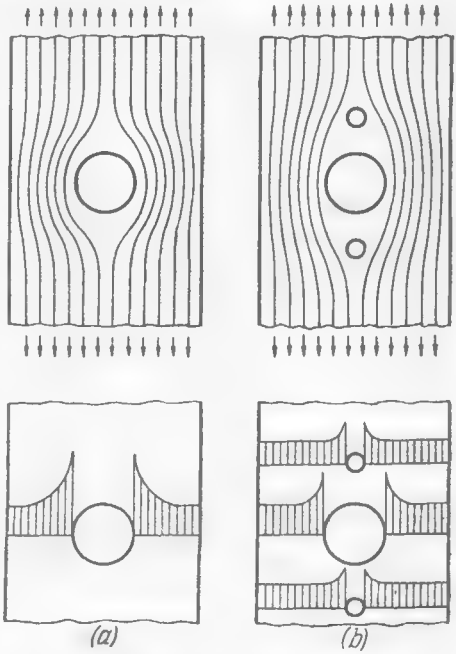


Fig. 195. Force flow in parts
(*a*) with a stress concentrator (hole); (*b*) with stress deconcentrators

Naturally, stress peaks occur around deconcentrators, however, it is evident that the height of peaks both near the chief stress raiser and deconcentrators is less than in the case of one stress raiser.

The position of the deconcentrators is of decisive importance. They are useful only when they make for the straightening of the force lines; otherwise they increase the force flux curvature and make the stress concentration still more intensive.

This can be easily seen in Fig. 196. With correctly positioned notches (Fig. 196*b*) the deconcentrators make the force lines more smooth, as if taking the material of the steps outside the force flux

area. In essence they have the same effect (but only weaker) as would have an improved smoothness of the transition section between the steps and specimen body, as is shown in Fig. 196*b* by dashed lines.

Incorrect position of additional notches (Fig. 196*c*) produces further force lines distortion and, hence, larger stress concentrations.

Another technique used to minimize the harmful effects of concentrators consists in cold reduction of the material on the surfaces of the concentrators and in the sections close to them. Thus, hole walls are made more dense by furnishing, while the surfaces around the hole are embossed; round fillets and annular recesses are rolled over. These procedures are aimed at producing in the material residual compressive stresses. Adding to the local tensile stresses

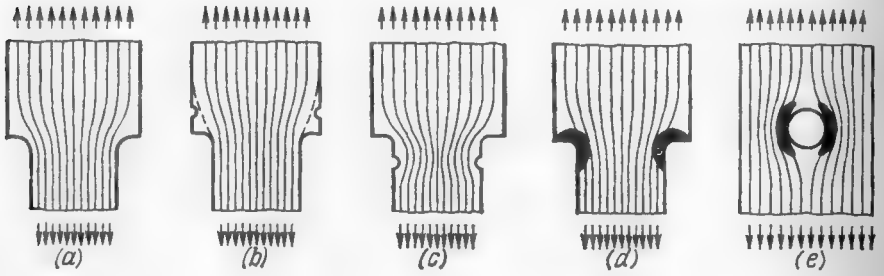


Fig. 196. Effect of additional notches on the uniformity of force flow

occurring in concentrators after the working load has been applied, such residual compressive stresses materially decrease the intensity of tensile stresses, thereby improving the strength at the areas of concentrators.

The above-described squeezings are misinterpreted as deconcentrators; their action is quite different. The purpose of deconcentrators is to straighten the force flow lines and that of squeezings—to strengthen the material. This difference is very important in practice since the arrangement of deconcentrators and squeezings is not the same: the former are positioned in the stream of force lines before the concentrator and after it, and the latter—at the focus of stress concentration (Fig. 196*d*, *e*).

The phenomenon of stress concentration caused by the factor of shape is aggravated in practice by the fact that the areas in which stress concentrators are located are almost always weakened by manufacturing conditions.

Accordingly, there are two types of concentrators: geometric concentrators (form concentrators) and manufacture concentrators.

In machined parts the weakening of transition sections is caused by cutting through the fibers which have been formed in the course of the hot working of blanks. In cast parts the transition sections

are usually weakened by casting defects which have been originated by structural distortions during crystallization of metal and solidification of ingot. The affected areas concentrate, as a rule, loose spots, voids, microcracks and internal stresses. In forged and stamped parts the transition sections have poor strength as a result of metal drawing in the areas.

(*j*) *Stress Concentrators*

Figure 197 shows schematically stress concentrators typical of such parts as plates, bars, etc. when the latter are subjected to tension-compression or flexure.

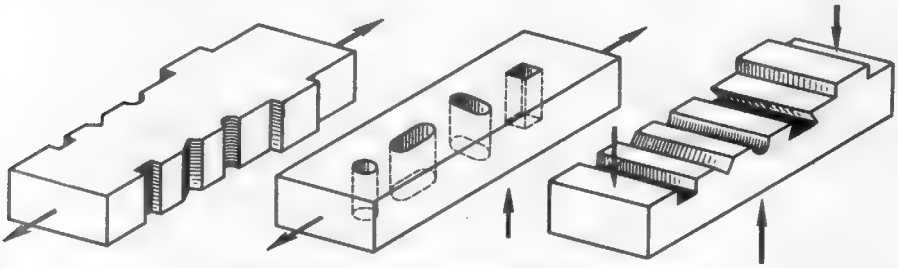


Fig. 197. Stress concentrators in flat bars

Stress concentrators typical of cylindrical parts (e.g., shafts) are cited in Table 22.

Stress concentrators on insides of hollow shafts are depicted in Fig. 198.

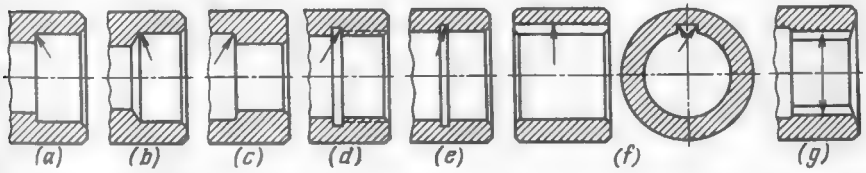


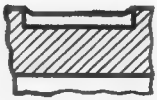
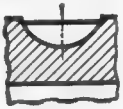
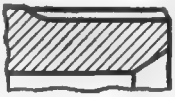
Fig. 198. Stress concentrators in hollow shafts (indicated by arrows)

Stress concentrations can also be originated by the internal defects of material: microvoids, microcracks, flakes, fine flaws, non-metallic inclusions (oxides, silicides), etc.

Concentration of stresses can be caused not only by the form of a part but also by the action of its mating part. An example is given in Fig. 199, which shows the stress distribution in the body of a coupling bolt obtained by experiment. Owing to the bolt shape, the

Table 22

Stress Concentrators

Sketch	Concentrators	Sketch	Concentrators	Sketch	Concentrators
	Shallow drilled holes		Annular recesses		Key-seats
			Grooves		
			Sharp-angled steps		Splines
	Holes		Under-cuts		Tooth spaces
					End-face slots
	Threaded holes		Flats		Welds
			Threads		Marks

highest stress is found at the transition position where the bolt stem meets the head and is three times greater than the mean stress σ_0 in the stem. Still much greater stress differential occurs in the plane of the nut end face ($\sigma_{max} = 5\sigma_0$).

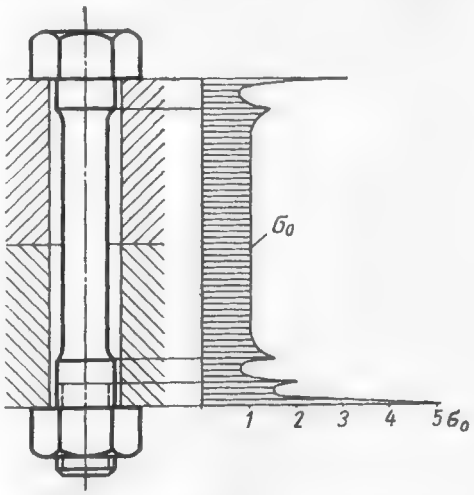


Fig. 199. Concentration of stresses in a bolted connection

(*k*) *Coefficient of Stress Concentration*

The enhancement of stresses at the areas of local weakenings is expressed in terms of the stress concentration coefficient.

There are two kinds of the coefficients: theoretical (found by methods of the mathematical theory of elasticity proceeding from an assumption of material homogeneity and ideal elasticity) and effective (found experimentally). The latter illustrates the properties of actual materials and some other factors which define actual increase of stresses.

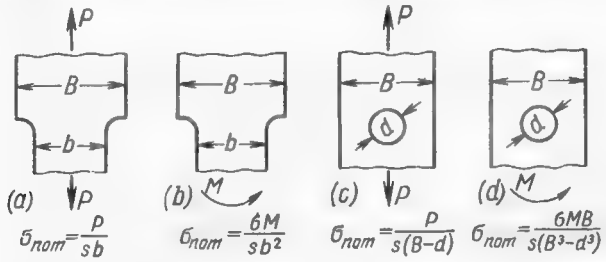


Fig. 200. Determination of nominal stresses
(*a*) and (*b*) in tension and in flexure for a stepped beam; (c) and (*d*) in tension and in flexure for a beam with a hole (*s*=beam thickness)

The theoretical coefficient of stress concentration

$$k_t = \frac{\sigma_{max}}{\sigma_{nom}}$$

where σ_{max} = maximum theoretical stress in the weakened area
σ_{nom} = nominal stress in the weakened area, calculated for the smallest section of the weakened area by the general formulae of the strength of materials (Fig. 200)

The theoretical coefficient of stress concentration has been defined for most forms of weakened sections found in practice (see, for example, Fig. 201).

The amount of increased stress in actual conditions can strongly deviate from theoretical values. It depends on the kind of loading (static or cyclic), type of material and its mechanical properties.

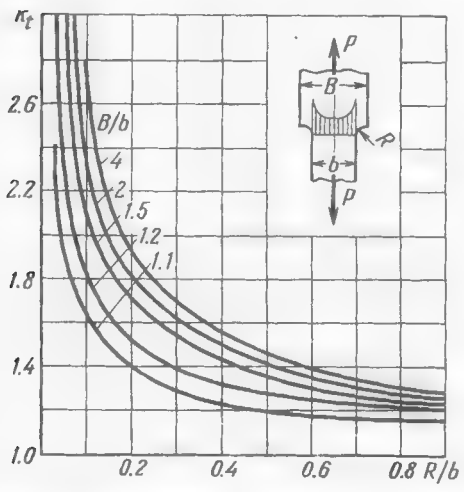


Fig. 201. Theoretical coefficient of stress concentration as a function of relative radius R/b at conjugated step sections (for a stepped beam in tension)

The actual amount of increased stress is evaluated through the stress concentration effective coefficient

$$k_e = \frac{\sigma'_{max}}{\sigma_{nom}}$$

where σ_{max}—maximum actual stress at concentration area.

With cyclic loads the stress concentration effective coefficient is deduced as a ratio between the fatigue strength σ of a plain specimen and the fatigue limit σ' of a specimen with a stress concentrator:

$$k_e = \frac{\sigma}{\sigma'}$$

Under static loading the amount of stress concentration will primarily depend on material plasticity. In plastic materials the phenomenon of stress concentration is rather vague. With the growth of stresses in the weakened zone the material begins to yield and a plastic hinge is formed there, which helps transfer forces to adjacent, less stressed areas.

For highly plastic materials the k_e parameter approaches to unity, which means that stress concentration does not occur.

In brittle materials the favourable yield effect is absent, so that higher stresses occur in weakened areas. As soon as local stresses pass beyond the ultimate strength of material, the latter undergoes brittle failure. For such materials the stress concentration effective coefficient approaches the theoretical coefficient ($k_e \approx k_t$).

There are, however, some exceptions. Thus, for instance, in the case of grey cast irons $k_e = 1$. This is attributed to the structural features of the material. Grey cast irons contain flaky inclusions of graphite which, owing to infinitesimal strength of the latter, act as equivalent internal notches and cause numerous local stress con-

centrations surpassing built-in stress concentrators (holes, recesses, etc.).

The phenomenon of stress concentration becomes more apparent under cyclic loading, this being attributable to the specific features typical of this type of loading. Plastic strains occurring as a result of stress concentrations follow one another at a high frequency and change their direction (under alternating loads), thereby gradually upsetting the material structure and leading to a fatigue failure.

The intensity of stress concentration will depend on material of a part, its chemical composition and homogeneity, heat treatment, mechanical strength of material, nature of a cycle, kind of concentrators, state of surface and absolute dimensions of a part.

Given below are approximate data which show how the material influences the effective coefficient of stress concentration:

Material	$\frac{k_e}{k_t}$ *ratio*
Alloy steels	0.85-0.9
Carbon and low-alloy steels . .	0.75-0.8
Wrought aluminium alloys . . .	0.7-0.75
Castable aluminium alloys . . .	0.6-0.65
Titanium alloys	0.55-0.6

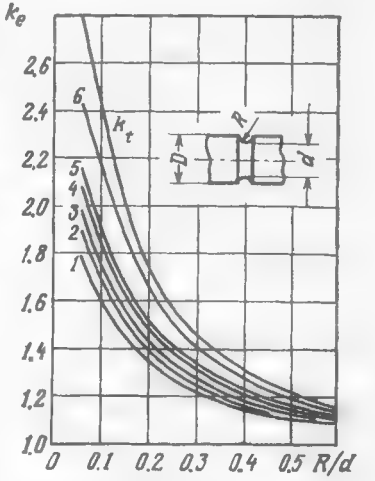


Fig. 202. Theoretical (k_t) and effective (k_e) stress concentration coefficients for bending of a cylindrical shaft having an undercut and made from materials of different strength (after Buchner)

1 — low-carbon annealed steel; *2* — low-carbon hot-rolled steel; *3* — medium-carbon normalized steel; *4* — medium-carbon structurally-improved steel; *5* — low-carbon work-hardened steel; *6* — structurally-improved alloy steel

As a rule, the lower the plasticity of the material and the higher its strength, the more extensive the stress concentration (Fig. 202).

However, the relationship between the stress concentration effective coefficient and material properties is very complex. Thus, for example, steels with martensite and troostite structure (hardened respectively with low and medium tempering) are less sensitive to stress concentration than refined and normalized steels with sorbite and sorbite-pearlite structures.

When taken within certain restricted limits, the correlation between the effective and theoretical coefficients of stress concentration can be expressed in the following form:

$$k_e = 1 + q\,(k_t - 1) \qquad (5.6)$$

where q = coefficient of material sensitivity to stress concentration.

In respect to high-strength alloy steels the q value approaches unity, while in relation to carbon and low-alloy steels this parameter varies within 0.6-0.8 (higher values corresponding to steels better in strength). For grey cast irons $q = 0$ ($k_e = 1$).

Figure 203 shows graphically the $\frac{k_e}{k_t}$ ratio calculated from Eq. (5.6) and taken in regard to different values of q and k_t.

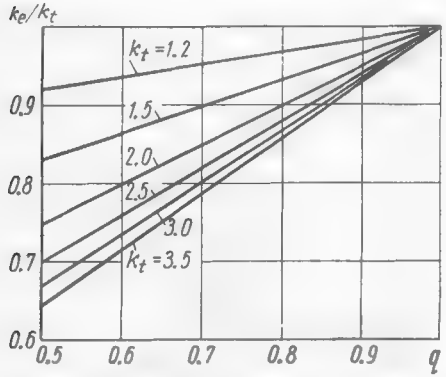


Fig. 203. Ratio of the effective stress concentration coefficient k_e to the theoretical stress concentration coefficient k_t as a function of the coefficient q of the sensitivity to stress concentration (for different values of k_t)

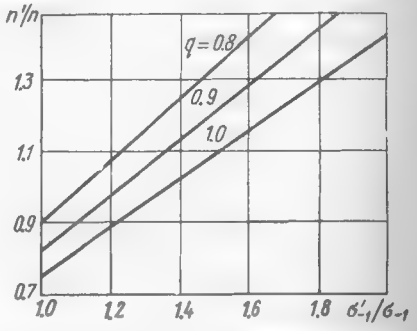


Fig. 204. Increase of strength margin when changing over to steels of higher fatigue strength with different values of sensitivity coefficient q. The value of q for the original steel q is taken at 0.6

Highe sensitivity of high-strength steels to stress concentration conceals their strength advantages. In many cases it is more advantageous to apply steels of moderate strength with lower values of the sensitivity coefficient.

Assume that a part is made of steel with moderate fatigue strength and with $q = 0.6$. Now let us determine the gain obtained by using steels with greater fatigue strength and correspondingly with higher q.

With geometric sizes of the part remaining the same, the nominal stresses σ_{nom} will also be the same. The relationship of the safety factor (determined in terms of maximum stresses σ_{max}) when changing to high-strength steels

$$\frac{n'}{n} = \frac{\sigma'_{-1}}{\sigma_{-1}} \frac{\sigma_{max}}{\sigma'_{max}} = \frac{\sigma'_{-1} k_e \sigma_{nom}}{\sigma_{-1} k'_e \sigma_{nom}} = \frac{\sigma'_{-1} k_e}{\sigma_{-1} k'_e} \tag{5.7}$$

where σ_{-1} and σ'_{-1} = fatigue limits taken respectively for original and higher-strength steels

k_e and k_e = effective coefficients of stress concentration, taken respectively for original ($q = 0.6$) and higher-strength steels

Let $k_t = 2.5$. The relation between k'_e (for different values of q) and initial k_e ($q = 0.6$) is presented graphically in Fig. 203.

The graph shown in Fig. 204 has been plotted with the aid of calculations based on Eq. (5.7). Here the variation of n'/n is given as a function of σ'_{-1}/σ_{-1} for different values of q.

As evident from the graph, the safety factor can appreciably be increased ($n'/n = 1.2$) when applying steel whose fatigue strength is at least 1.35 that of original steel when $q = 0.8$, 1.5 times when $q = 0.9$, and 1.7 when $q = 1$. With σ'_{-1}/σ_{-1} values below the above magnitudes, one may advantageously, as to the cost of manufacture, employ steels with a moderate strength.

The effect of the kind of loading on the magnitude of the stress concentration effective coefficient depends on the type of stress concentrator:

for notches and transverse holes

$$k_{e\ tens\text{-}compr} : k_{e\ flex} : k_{e\ tors} = 1 : 0.85 : 0.65$$

for fillets

$$k_{e\ flex} : k_{e\ tens\text{-}compr} : k_{e\ tors} = 1.2 : 1 : 0.8$$

Stress concentrations depend also on the kind of cycle. The higher the cycle asymmetry coefficient r and the greater the mean stresses σ_m, the sensitivity of q to notches and k_e is reduced.

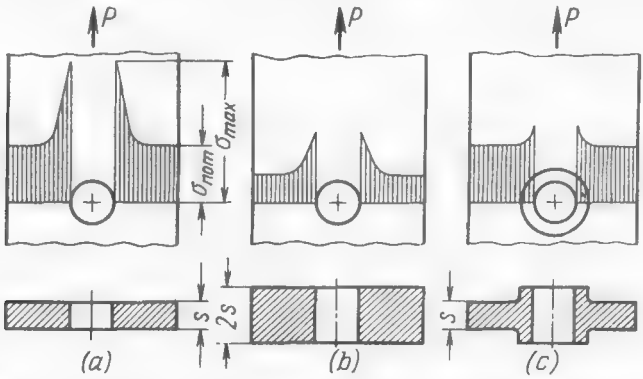


Fig. 205. Decreasing the maximum stress by lowering the nominal stress

The main influence on the stress concentration effective coefficient is given by the form of the weakened areas. Coefficient k_t, similarly to coefficient k_e, abruptly drops with the improvement in transition smoothness and lessening sectional gradient between non-weakened and weakened areas of the part (see Fig. 202).

Another way of increasing strength of weakened areas is reduction of *nominal* stresses at their concentration area.

Since $\sigma_{max} = k_e \sigma_{nom}$, the maximum stresses at the area of concentration (for a given k_e) depend on the value of σ_{nom}. Hence, all measures, contributing to the reduction of nominal stresses (larger sectional areas and higher moments of resistance where concentrators are positioned) lower the maximum stresses.

For a simple example we take a beam with a central hole (Fig. 205*a*). With k_e remaining constant it is possible to halve the maximum stresses by doubling the beam thickness s (Fig. 205*b*). Approximately the same result is obtained when increasing the beam strength locally with the aid of bosses positioned each side of the hole (Fig. 205*c*).

Strength of weakened areas can substantially be improved by strengthening processes. Some forms of processing (nitriding, shot-blasting, etc.) practically paralyse stress concentrations even in concentration-sensitive steels.

(*l*) *Scale Factor*

Fatigue strength becomes lower with increase in part dimensions. It is usual to characterize the influence of details' sizes by the *scale factor coefficient* ε_h, which is the ratio between the fatigue limit σ of a given specimen to the fatigue limit σ_0 of a laboratory specimen of small sizes (sectional diameter 5-10 mm)

$$\varepsilon_h = \frac{\sigma}{\sigma_0}$$

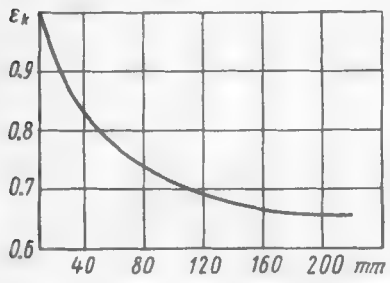


Fig. 206. Scale factor ε_h as a function of part diameter (average values after Lehr, Faulgaber and Peterson)

Figure 206 presents averaged values of the scale factor coefficient for structural steels.

The fatigue strength drops particularly sharply in the range of sectional diameters from 5 to 100 mm. With further increase of dimensions the scale effect is smoothed out.

The influence of dimensional characteristics on the fatigue strength of a rectangular section specimen from normalized carbon steel in the conditions of unilateral bending is illustrated graphically in Fig. 207. A specimen of the cross-sectional area $F = 60$ mm^2 and width $b = 11$ mm was assumed as a 100% strength reference piece. With the increase of the sectional area up to 10^4 mm^2 ($b = 140$ mm) the fatigue strength fell to 55% of the original value. The presence of weakening holes magnify the scale effect to a greater extent the less the hole diameter. With a hole diameter $d = 0.1\ b$

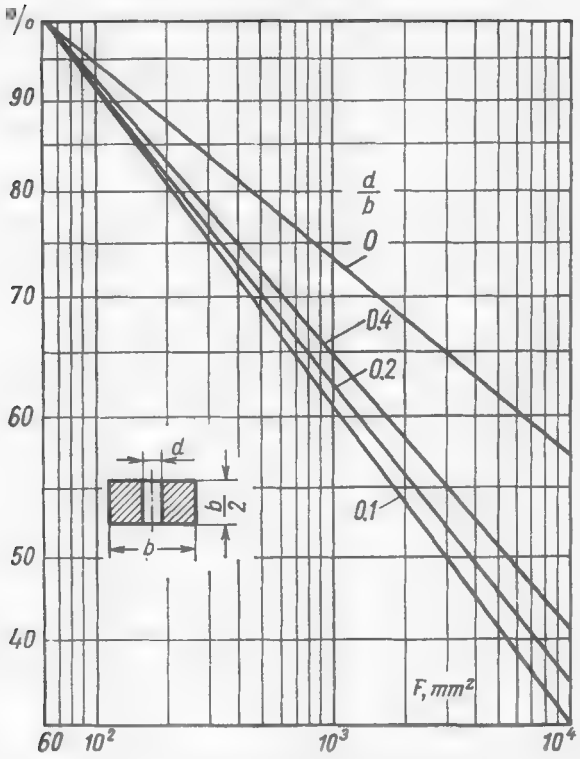


Fig. 207. Effect of size on fatigue strength (after Kermes and Büzek)

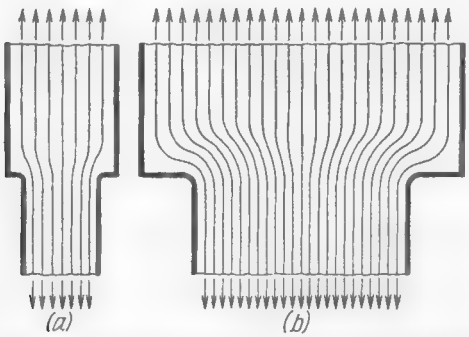


Fig. 208. Force flow lines in a stepped specimen

(*a*) small-size specimen; (*b*) geometrically similar larger specimen

the fatigue strength, when $F = 10^4$ mm^2, comprises only 30% of the original value.

Several assumptions have been put forward to explain the influence of dimensional sizes upon the fatigue strength of parts. According to statistical theory the enlargement in part size enhances probability of heterogeneities and internal defects.

From the technological point of view advanced in the first instance are the difficulties in obtaining uniform strength throughout the entire section of large parts, as, for example, during hot forming and heat treatment.

The increase of part size definitely makes stronger the influence of stress concentrations. This can be understood from the distribution of force flow lines in a stepped part subjected to tension (Fig. 208). Should the dimensions of a part be increased, while preserving fully the geometric similitude and with equal stresses (Fig. 208*b*), then the flow of force lines will change (with the same force line spacing): in the step zone the force lines curve more sharply than in the smaller part, thus indicating the enhancement of stress concentrations.

(*m*) *Surface Condition*

The fatigue strength strongly depends on the condition of surface, particularly in those cases of loading, when the severest stresses occur in subsurface layers (flexure, torsion and complex stressed states).

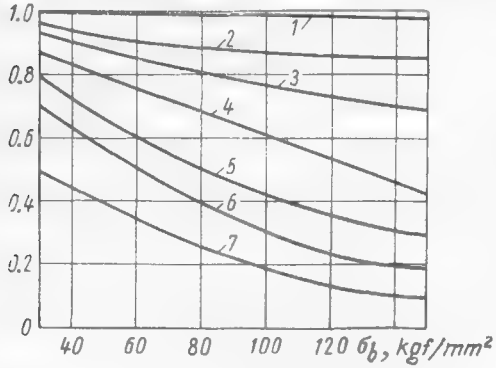


Fig. 209. Fatigue strength of steel specimens depending on the surface condition and tensile strength σ_b of the steel

1 — polished specimen; *2* — ground specimen; *3* — rough-turned specimen; *4* — specimen with a circular notch; *5* — specimen with scaly surface; *6* — specimen subjected to corrosion in fresh water; *7* — specimen subjected to corrosion in sea water

Rough machining, causing plastic shear, tears and microcracks in the surface layer sharply lowers the fatigue limit. Finish machining (polishing, superfinishing, etc.) improves fatigue strength. This feature is particularly expressed in small-sized parts.

Fatigue strength is adversely affected by chance scratches and damages to the surface layer and also surface wear. The fatigue strength is also badly affected by corrosion.

Figure 209 illustrates the fatigue strength of steel specimens subjected to various kinds of machining and surface damages as a function of tensile strength σ_b. For unity the fatigue strength

of a polished specimen of steel with tensile strength $\sigma_b = 30$ kgf/mm² was taken. From the graph it is seen that the influence of surfaces damages grows with the increase in material strength, which indicates the higher sensitivity of these materials to stress concentrations.

(*n*) *Other Factors*

Pressed, tapered and clamped connections with high compressive stresses on the bearing surfaces lower fatigue strength. Especially sharp is the fatigue strength drop in the compressive stress interval up to 4 kgf/mm² (Fig. 210). With further increase of pressure the effects of cold working decrease.

Galvanized metallic coatings negatively influence fatigue strength, particularly those of hard and durable metals (chrome, nickel). Coatings of plastic metals (copper, zinc, cadmium, tin, lead) have almost no effect upon the fatigue strength.

Reduction of fatigue strength, caused by electrolytic coatings, is mainly due to hydrogen embrittlement of metal — both of the part and coating.

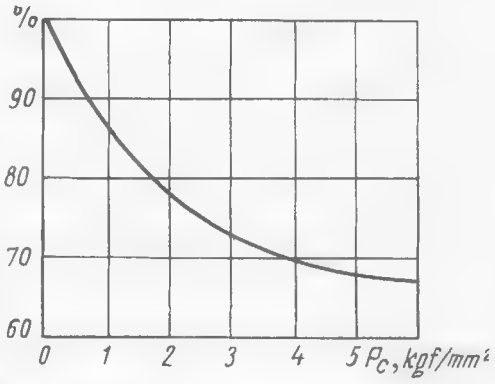


Fig. 210. Fatigue strength of press-fitted connections as a function of unit pressure p_c on contacting surfaces

In the course of electrolytic deposition the metal coating is saturated with hydrogen, acquiring a closely-packed hexagonal lattice typical of metal-hydrogen compounds. This induces considerable tensile stresses in the surface layer (6-15 kgf/mm² for chrome and nickel platings).

Furthermore, the fatigue strength of metal coatings is generally less than that of the metal part.

For all these reasons the initial cracks occur first of all, in the plating and then they propagate into the depth of the part.

Due to the yield in plastic metal coatings significant stresses do not occur.

The fatigue strength of chrome- and nickel-plated parts can substantially be increased by annealling at 400-500 °C: this lowers residual tensile stresses in the surface layer. The most efficient way to improve fatigue strength is to strain-harden the surface layer before coating and also to strain-harden the coating by shot-

blasting or rolling, which results in the inception of compressive stresses. Combined employment of the aforelisted technique will practically eliminate the weakening effects of electrolytic plating and even improve the fatigue strength as compared to the original value inherent in the material in the non-strained state.

(o) *Fatigue under Non-Stationary Loading Conditions*

Tests on fatigue according to Weller and damage tests to French are carried out under stable time conditions and continuously acting cyclic loads. This kind of load is typical of several machines, working continuously in a constant regime (stationary power motors, electric generators, machines built into automatic continuous production lines, etc.). The majority of machines operate in changing conditions with regular or irregular alternating cycles and different stress levels (transporting, construction, road-building and hoisting machines, machine tools, presses, hammers, etc.).

Depending on working conditions, the level of stresses can vary within broad limits (idle running, normal loading, overloads). In the course of operation some machines may be subjected to rather high overloads, which cannot only exceed the fatigue limits, but also often go beyond the material ultimate yield limit, thereby causing plastic deformations in the part.

Obviously, the fatigue limits, determined in tests under stationary conditions of cyclic loading are inapplicable to machines running under non-stationary conditions. The problem of fatigue strength under non-stationary working conditions is at present studied most attentively.

Non-stationary loading conditions may occasionally be schematically presented as operating in an alternating definite continuous group of cycles (block stresses) with rest periods (work — rest).

In the general case the conditions of non-stationary load are defined by the following parameters:

kind of load (tension-compression, bending, torsion); type of cycle (alternating symmetrical or asymmetrical loads, pulsating cycles);

number of cycles in a block (block size);

relative duration of work and rest periods;

regularity of stress changes within the block;

the value of maximum stress in a block and its relationship with the original fatigue limit under stationary conditions (degree of overload);

frequency of block (degree of load periodicity);

frequency of cyclic stresses within the block.

The combination of these parameters are named the load spectrum.

Some of several possible load spectra are presented schematically in Fig. 211. A special kind of non-stationary loading is presented by the block alternation of cyclic loads with periods of static loads (Fig. 211*i*, *j*).

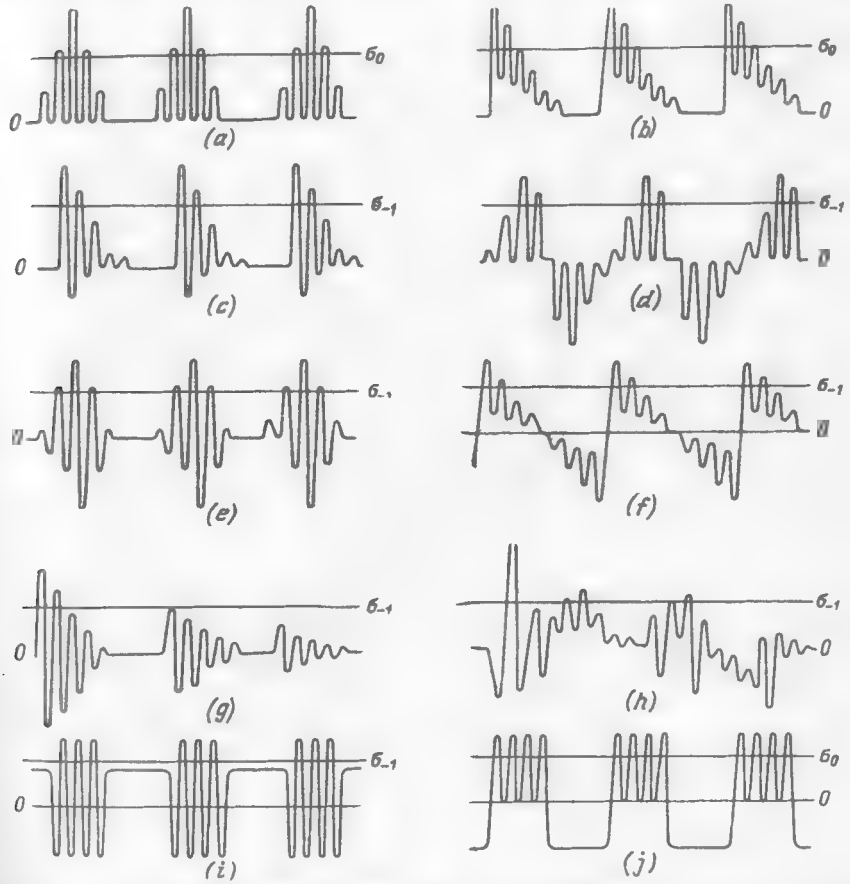


Fig. 211. Schematic stress spectra under non-stationary loading conditions (*a*) and (*b*) pulsating sign-constant cycles; (c) asymmetric alternating cycle; (*d*), (*e*), (*f*) and (*g*) symmetrical alternating cycles; (*h*) complex loading with random alternation of cycles; (*i*) and (*j*) combination of cyclic loading with periods of static loading

When testing under non-stationary conditions, the stress spectrum is given on the basis of possible or actual working conditions. The tests are undertaken at a changing value of some dominating factor (mostly the one, showing the degree of overloading). In this way a network of *secondary* fatigue curves is obtained, biased relative to the *primary* fatigue curve (i.e., a curve, typical of the stationary load).

Depending on the value of overloads, type of cycle, periodicity size of specimen and other factors, alternating loads may act either as strain-hardening or strain-softening ones, and have an effect on the secondary fatigue curves (Fig. 212). When secondary fatigue

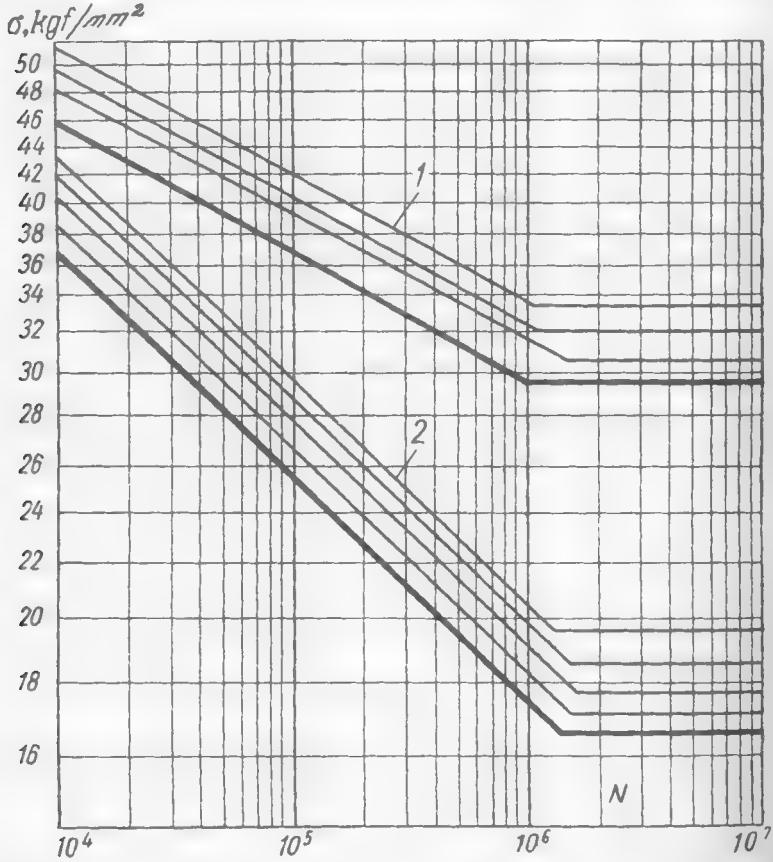


Fig. 212. Secondary fatigue curves in circular bending for various overloads (specimens made from steel grade 40)

Curves *1* — plain cylindrical specimens; curves *2*—specimens with annulas recesses. Heavy lines distinguish initial fatigue curves

curves are biased upwards and to the right this will signify strain-hardening of the material and enhancement of service life period within the limits of restricted durability. Opposite biasing (down to the left) indicates strain-softening of material and a reduction in restricted durability.

The fatigue strength limit is slightly increased when operating under short-time overloads about 1.5 those of the original fatigue

limit which alternate either with rest periods or with less strained loads.

In the majority of cases the increase of fatigue strength is the result of a reduced mean amplitude of stresses. Thermal rest of metal favourably effects the latter when operating under alternating work—rest periods. Heat, accumulated in microvolumes during work periods, dissipates during following pauses into adjacent cooler volumes so that an overstressed volume will meet the next cycle in a cooled, i.e., strain-hardened state.

The improvement of fatigue strength under the action of short-time overloads can be attributed to the strain-hardening, which occurs during plastic deformation of material microvolumes and is similar to cold working.

It has been established that irrecoverable strains bring about strain-hardening effects: randomization of crystalline lattices; thicker density of dislocations; refinement of crystalline clusters and still more extensive disorientation of the latter; displacement of grain boundaries; deformation of cleavage surfaces as a result of plastic shear reaching the grain surface, thereby reinforcing intergranular bonds. Elements C, O and N show poorer dissolubility in α-iron; these elements precipitate from solid solution, forming highly-dispersed carbides, oxides and nitrides, which occur as clouds arresting the extension of dislocations. In hardened steels the residual austenite decomposes and turns into finely acicular martensite typical of strain phase.

Growth of strength, observed when raising the overload to a known limit, can be attributed to the increased number of microvolumes, undergoing plastic strains, and also by the enhancement of strain-hardening intensity. After a certain stage the strain-hardening process discontinues. In fact, this situation arrives at such level and rate of stress alternation, when irreversible intra- and intercrystallite damages occur in the material, affecting the continuity of the latter.

The studies of fatigue strength under non-stationary conditions are of theoretical and practical importance as they allow a still deeper penetration into the nature of the fatigue phenomenon, more rational use of the material and a more accurate determination of a structure's service life under working conditions. However the calculations are very complicated. A large quantity of experimental data is necessary to explain the regularities of change in fatigue limits with different load spectra. Also to be duly accounted for are stress concentration factors, surface conditions, etc., since their influence upon the fatigue curves under non-stationary conditions may differ much from those under stationary loads and often significantly (Fig. 212).

Efforts have been made to determine the durability under non-stationary conditions using the Palmgren hypothesis on the cumulative summation of damages. A stress curve is subdivided into separate segments (steps) having approximately the same stress amplitude of stresses. Since the load character at separate steps may be different, the mean stresses at each step lead to stresses equivalent in terms of damaging action in the symmetrical cycle.

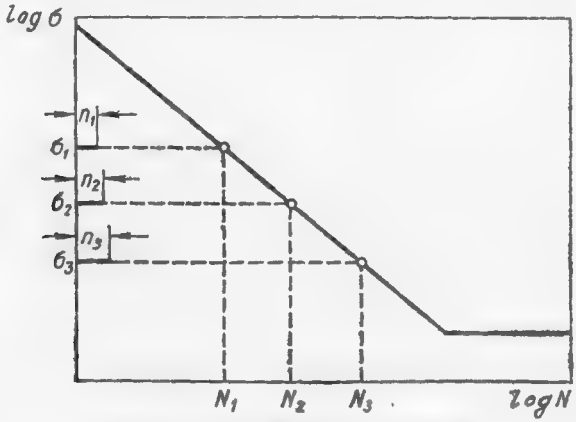


Fig. 213. Summation of damages

According to the Palmgren hypothesis the amount of fatigue damage is linearly dependent upon the number of cycles at a given stress level, thus comprising a proportional quota of the full damage (failure), ensueing at the durability, ultimate for the given stress level.

Let N_1 be the number of cycles to failure at a given level of stresses σ_1 (Fig. 213). Then, one cycle of stress σ_1 will cause damage equal to $1/N_1$, and n_1 of cycles—n_1/N_1 of the full damage. If cyclic durability under stress σ_2 is equal to N_2 of cycles, while the number of cycles for the given stress is n_2, then the fatigue damage will comprise n_2/N_2 of the full damage, etc.

Further it is assumed that accumulation of damages is not dependent on the order of the alternating steps.

The terms of damage summation are written as follows

$$\frac{n_1}{N_1}+\frac{n_2}{N_2}+\frac{n_3}{N_3}+\ldots=1$$

where the full damage (failure) is taken as unity.

Dividing both halves of the equation by durability N corresponding to failure gives

$$\frac{n_1}{NN_1}+\frac{n_2}{NN_2}+\frac{n_3}{NN_3}+\ldots=\frac{1}{N}$$

Here $\frac{n_1}{N} = \tau_1$; $\frac{n_2}{N} = \tau_2$; $\frac{n_3}{N} = \tau_3$ . . . —duration of stress action at cycle steps in relation to the expected limit cycle duration. Substituting these values into the previous expressions, one will obtain

$$\frac{\tau_1}{N_1} + \frac{\tau_2}{N_2} + \frac{\tau_3}{N_3} + \ldots = \frac{1}{N}$$

In the generalized form (Minor's equation)

$$N = \frac{1}{\sum_{i=1}^{i=r} \frac{\tau_i}{N_i}}$$

where τ_i = relative durations of acting stresses actions on the cycle steps

N_i = durabilities, corresponding to the levels of these stresses

r = total number of steps

Experiments do not prove this relationship. According to experimental data

$$N = \frac{0.5-2}{\sum_{i=1}^{i=r} \frac{\tau_i}{N_i}}$$

The variations are so great that the possibility of using Minor's equations in calculations is questionable. The Palmgren theory is based upon rather primitive assumptions far from true physical nature of the phenomena. In particular, it does not consider the actual kinetic development of damage with the increase of a number of cycles, influence of the stress level upon the magnitude and kinetics of damage, influence of the alternating steps on the damage, rest periods between steps and blocks of steps.

At present the philosophy of fatigue strength (particularly strength under non-stationary conditions) is most strongly influenced by the science of physics of metals. If not accompanied by the creation of an orderly physical theory of metal fatigue strength, the empirical study of this phenomenon will boil down to a mere accumulation of statistical data, suitable for use in particular engineering calculations.

A problem also emerges how to use most rationally the observed regularities of strain-hardening in order to improve endurance and durability of structures. The problem consists in the development of rational training of parts through higher cyclic loads, alternating with intervals of rest. In addition it is necessary to develop techniques of strain-hardening of parts through dosed plastic deformation by applying static and cyclic loads.

It has been established that fatigue limit of specimens, loaded with cyclic tensile stresses, is substantially increased by preliminary deformation as this work hardens the material (Fig. 214). The effect of plastic deformation is particularly prominent when the deforming load and the working load have the same sign.

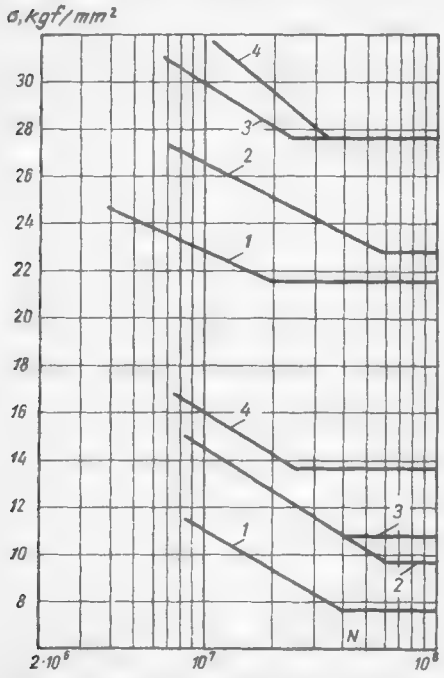


Fig. 214. Effect of preliminary deformation of the fatigue strength of specimens 14 mm in diameter and made from steel grade Ст. 7 (upper curves are for plain specimens; lower curves, for specimens with a stress concentrator — an annular recess)

1 — without deformation; *2*, *3* and *4* — with torsional deformation of 25, 50 and 75%, respectively (after N. V. Kudryavtsev and V. V. Rumynova)

Deformation also causes residual stresses in the surface layers of opposite sign to those of working stresses. This feature is now utilized in the process of spring presetting and can be applied for strengthening other parts, e.g., shafts subjected to a circular of repeated unilateral bending.

An even greater effect is obtained by combining zonal plastic strain with outside cold working of zones being deformed (strain-hardening in a stressed state).

(*p*) *Fatigue Limits of Parts*

When generalized, the fatigue limits can be expressed in the following form

$$\sigma_{D\,part} = k_1 k_2 k_3 k_4 k_5 k_6 k_7 k_8 \frac{e_k}{k_e} \sigma_D$$

where σ_D = fatigue limit of a plain polished specimen of a given material at a given kind of strain-hardening process and type of loading

k_1 = coefficient of surface finish

k_2 = coefficient of corrosion attack

k_3 = coefficient, accounting for surface damage due to wear in the process of service

k_4 = coefficient, accounting for cycle frequency factor

k_5 = coefficient, accounting for the degree of load impact

k_6 = coefficient, accounting for thermal conditions under which the part operates
k_7 = coefficient, accounting for heterogeneity of material and scatter of strength characteristics
k_8 — coefficient of load spectrum
ε_k = coefficient of scale factor
k_e = stress concentration effective coefficient

The k_1, ε_k and k_e coefficients can be inferred from available experimental data, k_8 — from test data at a given load spectrum. In usual circumstances the influence of the k_4-k_6 coefficients are insignificant. The other uncertainties are generally accounted for through the factor of safety, taken as equal to 1.5-2.

Much more reliable is the full-scale test of a part under the conditions, which most closely approach the actual working conditions and load spectrum. In this the specific features of the part design are directly considered. The factor of safety must include the scatter factors of material characteristics, damages inflicted in the process of work and also deviations of actual loading conditions from those during the test.

5.1. Improvement of Fatigue Strength

Up to now the physical phenomena of fatigue strength, have still not been studied to the degree which allows orderly fatigue strength calculations of parts to be undertaken. The lack of fundamental physical principles forces one to accumulate experimental data which does not always give true answers. The situation is aggravated by the fact that the data, being supplied by various experimenters, have extensive scatter and frequently due to the differences in the testing techniques are incompatible and even contradictory. Calculating formulae become more complicated because of the different levels of fresh data, introduction of correction coefficients and also the varied subjects of registered factors.

In these conditions comprehension of general regularities which determine the fatigue strength of parts is of very great importance. The designer has at his disposal enormous possibilities of improving fatigue strength. Sensible design, based on knowledge of fatigue strength regularities at times gives rather more than formal calculations and avoids mistakes that would otherwise have to be rectified, for instance, through strengthening processing.

The designer must know and confidently apply his own technological and constructional recommendations on methods of improving fatigue strength.

In many cases one can eliminate the source and achieve if not full exclusion of cyclic loads then at least their partial elimination. Even in machines having definite cyclic action it is possible to lower

significantly the maximum values of cyclic stresses and their amplitudes and also mitigate impact loads.

One of the principal ways is to enhance elasticity of parts in the direction of acting load and introduce elastic links between parts transmitting and receiving loads.

For example, increasing the elasticity of bolts in joints subjected to cyclic loads reduces both the value of acting loads on the bolts and the interval between the maximum and minimum loads, i.e., favours the improvement of fatigue strength.

During cyclic torsion peak stresses may be cushioned, consequently decreasing cyclic amplitude, by using flexible couplings between members accepting the torque. For example, the provision of spring dampers between shafts and gears will lower peak stresses in teeth and make gears run more smoothly and quietly.

The change-over from rolling contact bearings to plain bearings in conrod-crank mechanisms lowers peak loads as the oil film has a damping action. The work being spent for displacing oil from the bearing clearance absorbs the impulse of the acting forces, which helps to reduce the loads imposed on the mechanism.

Cyclic loads, arising in shafts, may in some cases (non-driving gear wheels, idlers) be eliminated by mounting rotating parts on axles.

In many cases the onset of high alternating loads is associated with the appearance of resonance vibrations in parts of the mechanism. This dangerous kind of cyclic loading can be removed by employing spring, pendulum, hydraulic or friction type dampers.

Vibrations of machines and units, presenting sources of alternating loads, are eliminated or cushioned by suspending them on anti-vibration or vibration-damped mountings.

Sometimes cyclic loads can be completely or almost completely removed by means of more accurate manufacture of parts and their supports. The elimination of static and dynamic disbalance in high-speed rotors, causing alternating loads in supports and housings, is an example of this. Better teeth manufacture, smaller errors in pitch and tooth-thickness, more precise profiles, reduced pitch circle runout, accurate gear helices, etc., will remove cyclic loads being caused by such errors.

It is very important to note that any measure, aimed at decreasing nominal value of stress simultaneously improves fatigue strength. Belonging to such measures are rational spacing of supports, elimination of irrational loads, increasing of cross-sections at parts subjected to cyclic loading, enlarging of surface contact areas (with cyclic contact stresses).

All rational design rules being applied to statically loaded constructions not only remain valid but are also of significance for cyclically loaded structures.

In the cases when liquidation of cyclic loads or at least a reduction in cyclic stresses is not successful, special methods of improving fatigue strength should be used. These techniques can be divided into process and constructional. In the first case strengthening is attained through special processing, while in the second case by imparting shapes which are most rational to fatigue strength.

(*a*) *Technological Methods of Improving Fatigue Strength*

Extremely heavy influence on the fatigue strength value is exerted by the steel melting process. Vacuumized steels have improved fatigue strength qualities. The same is true for steels, produced by electric remelting in a vacuum or under a layer of synthetic slag. A substantial improvement in fatigue strength is also assured by thermal-mechanical processes (particularly LTTP).

Thermal and chemical-thermal treatment of steels have a favourable effect, especially when stress raisers are present. According to their influence on the fatigue strength these processes can be arranged approximately in the following order: structural refinement and normalizing, low-temper hardening, case-hardening, induction hardening, cyaniding and nitriding.

The strain-hardening in the latter four cases is dictated in the main by the appearance of compressive stresses (of the order of 40-80 kgf/mm^2) in the surface layer due to a larger unit volume forming in the structure (martensite—in case-hardening and induction hardening, nitrides and carbonitrides—in cyaniding and nitriding) which is bigger than the basic one.

Creation of prior compressive stresses is equivalent to a reduction of mean stresses in tensile load cycles and increased asymmetry cycle coefficient which as easily seen from the Smith's diagram leads to a higher fatigue limit.

Moreover, increased surface hardness, obtained by thermal and chemical-thermal treatment, averts loss of strength due to wear, accidental scratches or some other defects.

Optimum surface layer thicknesses: in case-hardening, 0.4-0.6 mm; cyaniding 0.15-0.2 mm; nitriding, 0.3-0.5 mm; induction hardening, 2-4 mm. It is essential that strain-hardening processes involve all parts on the surfaces with stress concentrators (Fig. 215*a*, *b*) since substantial tensile stresses may appear at the boundaries of treated and non-treated zones, these stresses being extremely dangerous if located in the stress concentration zone.

Nitriding is the most effective since it practically eliminates the influence of stress concentrations. For nitrided parts the coefficient q, showing sensitivity to stress concentration, is close to zero (i.e., the effective coefficient of stress concentration $k_e \approx 1$).

Since no appreciable changes in dimensions and shape of a part are caused by nitriding, it can be employed as a final stage manufacturing process. This eliminates the final grinding process, which affects the strength due to the defects inherent to that operation (burnings, cracks, etc.). Combination of a high hardness, good fatigue strength and corrosion-resistance makes the nitriding treatment extremely valuable in processing the parts expected to be used under high cyclic loads and wear (crankshafts, heavily loaded gears, etc.).

The surfaces of parts working under high cyclic loads must be machined to the maximum economically acceptable surface finish standards. All kinds of finishing operations (polishing, lapping, superfinishing, etc.) smoothing down micro-roughness which remains on the surface after previous coarse machining operations contribute to fatigue strength, especially for details made from high-strength and hard materials.

Very good results are achieved by polishing under pressure, which affects deeply the structure of surface layers. Crystallites, separated during preceding machining, close together and are united by the action of pressure and heat evolved in friction. The surface layer is thus "compacted". Sharp edges of microirregularities are smoothed down, while micro-valleys and microcracks are levelled up. The smoothed surfaces of parts offer better corrosion-resistance.

For this reason all important components, subjected to high cyclic loads, are polished all-round not only on fitting and friction surfaces.

Moreover, it is easier to detect defects such as flakes, flaws, heat-treatment cracks, etc., on polished surfaces.

Press polishing is accomplished under 1-2 kgf/mm^2 pressure, using laps in the form of bronze or cast iron shoes, whose working surface is charged with micropowders (flours) of abrasive matter (carborund, boron carbide, diamond dust). Superfine polishing (buffing) is carried out with the aid of soft laps (babbit, wood, leather, suede, felt) employed in combination with lapping pastes of ГОИ type. Final finishing is done without any abrasives, using only kerosene or solvent naphta lubricants.

Excellent results have been obtained when polishing rubbing surfaces under pressure with colloidal graphite or (which is still better) molybdenum disulphide: this process improves wear-resistance and imparts better corrosion-resistance.

The principal methods of strengthening by plastic surface layer deformation are shot-blasting, rolling and caulking (cold calibration stamping); holes by expansion, ball sizing, piercing with smoothing broaches (driftpeening). Applicable also are diamond smoothing tools (i.e., hardening the surface with a rounded diamond tool), hardening by turning with carbide-tipped tools having large negative rake angle; caulking by explosion or electrohydraulic shock; abrasive-jet

polishing; ultrasonic hardening; hydraulic-pulse hardening with a high-pressure jet (10-20 thousands of atmospheres).

Exceptionally good results have been reached on surfaces, which were case or induction hardened prior to the cold working procedure. Hardening effects of plastic strains in this case can be attributed not only to the inception of compressive stresses, but also due to structural changes occurring in the material surface layer as a result of work hardening (the residual austenite decomposes and highly dispersible martensite emerges). Regardless of the kind of strain-hardening applied, it is necessary to induce in the surface layer stresses, which exceed the material's yield limit under volumetric compression.

Hardening through plastic deformation is practised mostly for steels. Brittle materials, for example, grey cast irons, are less prone to this type of structural improvement. Cold work hardening can also be applied to ferrite and ferrite-pearlite cast irons, even though the effect of strain-hardening is weaker in this case when compared with plastic materials. Other kinds of cast irons hardenable in the above-said way, are malleable cast irons, as well as modified and high-strength irons with globular graphite.

Shot blasting is one of the simplest and most universal of modern strain-hardening techniques. When chosen correctly, this procedure enables fatigue strength to be enhanced 1.5-1.6 times. However, it slightly impairs surface quality therefore precision surfaces should be carefully finished after being shot-blasted.

As the optimum schemes of hardening by plastic deformation are insufficiently developed, their use is determined experimentally for each individual case.

The shot peening procedure is selected to suit the properties of the material, its hardness and strength. Overhardening will readily cause brittleness and hairline imperfections in the surface layer. Approximate shot blasting recommendations for steel products: velocity of shot—50-60 m/s, flow rate 50-80 kg/min, angle of attack (jet incidence relative to component surface) 60-90°. Duration of process 2-5 min. Shot diameter (hardened steel balls)—0.5-1 mm.

When rolling the surface, the pressure force P, exerted onto a roll is chosen so as to produce in the surface layer stresses (to Hertz) amounting to 350-600 kgf/mm^2. An empirical formula for the force

$$P = 3 \cdot 10^{-3} \sigma_{0.2}^2 DB \frac{a}{1+a} \text{ kgf} \qquad (5.8)$$

where D = component diameter, mm
$\sigma_{0.2}$ = material yield limit, kgf/mm^2
B = usable width of a roll, mm
$a = \frac{D_{roll}}{D}$ = ratio between diameters of roll and component

In the range of most widely applied values $a = 0.7\text{-}1.0$, the value $\frac{a}{1+a} \approx 0.5$. Substituting this value into Eq. (5.8)

$$P = 1.5 \cdot 10^{-3} \sigma_{0.2}^2 DB \text{ kgf}$$

Generally roll width $B = 10\text{-}20$ mm. The component surface speed 10-20 m/min. Longitudinal feed $s = (0.05\text{-}0.1)\,B$, on average 0.5-1 mm/rev. Number of passes 1-3.

In order to relieve pressure, exerted upon a roll, vibrorolling has become very popular, in which a roll is imparted intermittently radial displacements from a mechanical or air-operated actuator.

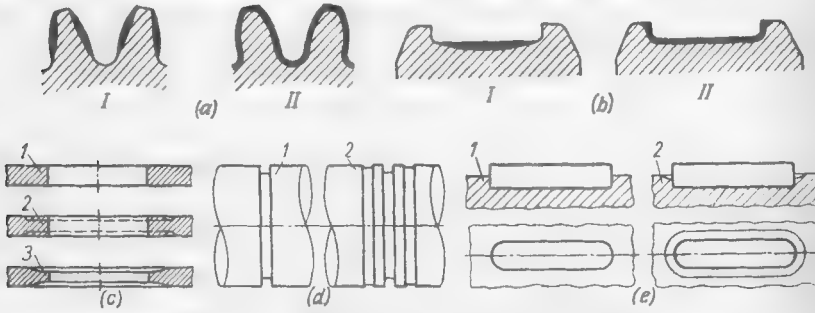


Fig. 215. Enhancement of fatigue strength

(*a*) and (*b*) surface hardening: *I* — incorrect; *II* — correct; (*c*) all-round swaging of hole edges; (*d*) rolling of annular grooves; (*e*) embossing around a keyway; *1* — original part; *2* and *3* — strengthened parts

When the rolling conditions have been correctly chosen, the residual compressive stresses in the surface layer will comprise 70-80 kgf/mm^2. The depth of the strain-hardened layer will reach 0.1-0.5 mm.

In contrast to shot peening, which slightly worsens the original surface finish, rolling improves it.

The finer the preceding machining the more favourable the rolling effect. To obtain the best results, it is better to finish machine (even polish) the component surface prior to rolling.

As component dimensions are hardly changed by rolling, the latter may be regarded as a final manufacturing operation. The possibilities of size change or final lapping after work hardening is not excluded: experience proves that removal of a work-hardened layer to the depth of 0.05-0.1 mm (the total thickness of the layer being 0.3-0.5 mm) brings about no appreciable decline of hardening effects.

It is useful to subject strain-hardened parts to stabilizing treatment, by heating to 100-150°C. At higher temperatures the residual compressive stresses in the hardened layer reduce and at 400-500°C vanish completely.

Rolling of threads improves their strength 1.5-2 times, and practically eliminates stress concentrations at thread roots, this being of particular merit when dealing with stress concentration sensitive steels.

Plastic deformation is widely used for strengthening hole edges punched in sheet metals. Hole edges are stress concentrators, due to plastic displacements, tears and microcracks which occur under the action of a punch. Circular reduction of hole edges (Fig. 215*c*) to a significant measure liquidates their weakening effects. Shafts, weakened by undercuts, are work hardened by rolling adjacent areas (Fig. 215*d*).

To improve strength of shafts weakened by keyways, the keyway is caulked around the key introduced into the way (Fig. 215*e*). Apart from reduced stress concentration, this measure also adds to the key fitting strength.

A markedly improved fatigue strength is displayed by elements which have been obtained by direct squeezing (rolling of threads, gear teeth and splines). In these cases the metal fibres are not cut as when machining large solid blanks, but are forced into the required configuration by the pressing process.

Normally threads are cold rolled, but gear teeth and splines are first hot rolled and then dimensionally sized cold.

5.2. Design of Cyclically Loaded Components

(*a*) *Reduction of Stress Concentrations*

The reduction of stress concentrations—is one of the major means of increasing fatigue strength.

Sometimes stress concentrators cannot be removed entirely. Under such circumstances the designer should do his best to substitute sharp concentrators with moderate ones. Example: screwed holes, belonging to most intensive stress concentrators, must, whenever feasible, be replaced by plain holes, whose negative effect is less and can be brought still lower by some additional measures.

Stress concentrators should be removed from most stressed zones of a part and disposed (if permitted by the design) in the zones of least stresses.

In order to decrease nominal stresses it is better to enlarge those cross-sections where stress concentrators are located.

Some examples of eliminating and decreasing stress concentrations are illustrated in Table 23.

Fillets. Concentration of stresses in entering angles of stepped parts (e.g., stepped shafts) can substantially be reduced by imparting rational forms to the conjugated steps.

Table 23

Removal of Stress Concentrations

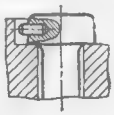
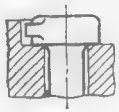
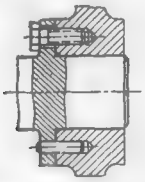
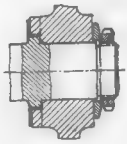
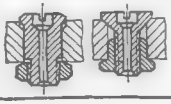
Original design	Improvement	Essence of improvement
Bolt head The locking lug is arranged in the most stressed section, thus causing sharp stress concentration. It also reduces the area of the critical section		The lug is fitted in the bolt head. Nominal stresses reduced, but stress concentration still remains
		The lug is now an integral part of the bolt head. Stress concentration eliminated
Mounting of a slip-on rotor upon a shaft  Fastening bolts and locating pins enter the hub body, the most stressed part of the rotor		The rotor is clamped with a circular nut. Torque is transmitted by teeth in the hub face which fit into slots on the shaft flange
Securing a plug in a shaft Thread causes stress concentration		Threads removed, press-fit substituted
		Threads removed, expended fitting substituted
		Direct thread connections to shaft eliminated. Nut and bolt form of fitting substituted
		Screwed joint eliminated. And fastening with threaded hole substituted

Table 23 (continued)

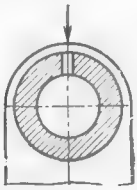
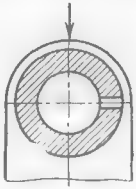
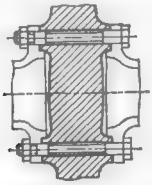
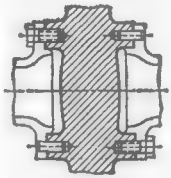
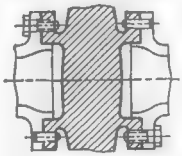
Original design	Improvement	Essence of improvement
Oil hole in crankshaft rod journal  Hole positioned at the part of maximum bending stresses, occurring due to ignition		Hole transferred to neutral zone (hole position must be correlated with load vector diagramme)
Fixing a solid turbine rotor to a divided shaft  Fastening bolts, passing through the rotor, sharply weaken it		Bolt receiving holes positioned in thickened annular bosses on the rotor arranged beyond stressed sections
		Bolt receiving holes positioned in flanges brought from the rotor body
Securing an antifriction bearing Thread weakens the loaded part of the shaft		Threaded collar is moved from the middle part to an unloaded shaft end face

Table 23 (continued)

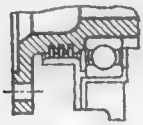
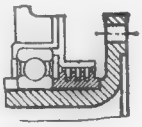
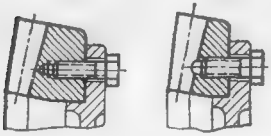
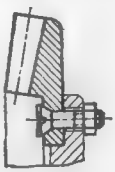
Original design	Improvement	Essence of improvement
Sealing unit with split spring rings  Ring receiving grooves weaken the shaft		Grooves are machined in a separate detail
Two concentrators (an external step and an internal recess) positioned in the same plane. The stress fields, being produced by the concentrators, accumulate. Reduced cross-sectional area increases nominal stresses		Stress concentrators positioned in different planes
Securing a bevel gear rim to a disk  The stresses, caused by bolt holes, add to the stresses near tooth root		Holes are repositioned away from gear teeth on an extension disk
Bevel gear rim Two stress concentrators combine (i. e. tooth gashes and sharp face edges)		Gear rim extended and strengthened. Smooth transitions introduced at rim-teeth conjugation areas

Table 23 (continued)

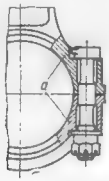
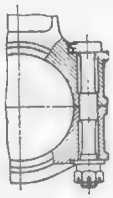
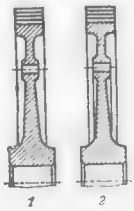
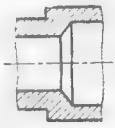
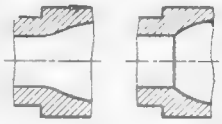
Original design	Improvement	Essence of improvement
Screwed rod		Screwed portion enlarged
Conrod big end  At positions *a* the part is weakened by bolt head and nut recesses		Sectional areas of weakened portions enlarged
Turbine rotor Rotor disk weakened by relieving holes		1. Holes strengthened with bosses 2. Holes positioned in a strengthening ring
Hollow shaft		Internal stress concentrator moved
Two concentrators combine (external and internal entering angles)		Internal angles given smooth streamlined forms

Table 23 (continued)

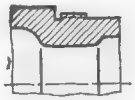
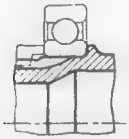
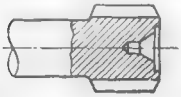
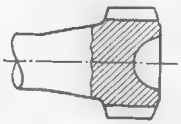
Original design	Improvement	Essence of improvement
Threaded shaft		Thread portion made thicker
Bearing installation on splined shaft  Three stress concentrators combine (i. e. entering angle, splines and internal steps)		Shaft cross-sectional areas with stress concentrators enlarged; internal surfaces streamlined
Bearing installation on shaft Shaft weakened by circlip grooves		Shaft strengthened at weakened portions
A splined spring shaft  Stress concentration at spline roots		Shaft strengthened at stress concentration areas

Table 23 (continued)

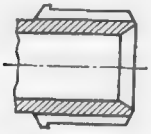
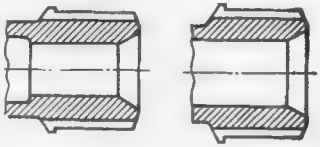
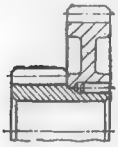
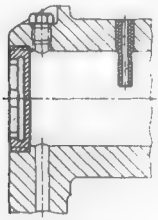
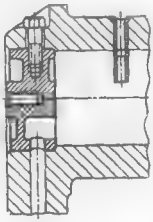
Original design	Improvement	Essence of improvement
Shaft with external splines  Stress concentration at spline roots		Shaft strengthened at weakened areas
Gear-to-shaft joint Stress concentrations at dowel positions		Dowel spacing diameter increased (i. e. reducing forces, acting upon the joint). Shaft and hub made thicker
Cluster gear  Stress concentration at dowel positions		The large gear fitted onto extended teeth of the small gear. Shaft and hub sections, transmitting torque are made thicker

Table 23 (continued)

Original design	Improvement	Essence of improvement
Crankshaft neck with an oil delivery system to crankpin bearing  The end-cap, plug and an oil inlet pipe are threaded: thread causes stress concentrations		The end-cap is slide-fitted in the shaft and locked by a bolt used instead of the threaded end-cap; the oil inlet pipe is extended through the end-cap into the shaft bore

Acute entering angles at a transition section (Fig. 216*a*, *b*) cause sharp stress concentrations. Tapered conjugations (Fig. 216*c*) add to the strength of transitional areas, but shorten the cylindrical

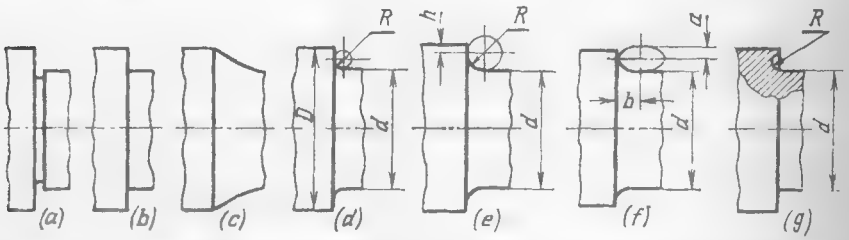


Fig. 216. Decreasing stress concentrations in the entry angles of stepped shafts

surface length of the smaller diameter. That is why such conjugations are applicable only at free transitions when this part of the detail does not mate with adjacent parts.

Most often stress concentrations at transitional areas are reduced by fillets (Fig. 216*d-g*).

The effectiveness of a fillet depends on its radius (Fig. 217).

Stress concentrations fall with reduction in diameter gradient and with an increase in ratio ρ. Significant gains in strength are

obtained with ρ value approximately equal to 0.1 for large diameter gradients and when ρ = 0.05-0.08 for small diameter gradients.

The maximum relative radius of a fillet, equal to $\rho_{max} = 0.5\left(\frac{D}{d} - 1\right)$ is limited by the diameter gradient. Usually in practice $D/d \approx 1.2$, so that the relative fillet radius cannot exceed 0.1. If a component fitted on the shaft has a rectangular supporting face at its step (*h*, see Fig. 216*e*), the radius must be even smaller.

Elliptic fillets (see Fig. 216*f*) at the same diameter gradient assure a comparatively greater (approximately by 20%) strength. The effectiveness of such fillets depends on the ratio of the ellipse major semiaxis *b* to the shaft diameter *d*.

When $b = (0.4\text{-}0.45)\,d$ and $\frac{a}{b} = 0.4$, the stress concentration coefficient will not exceed 1.5.

The disadvantage of elliptic fillets is the shortening of the cylindrical portion of a shaft, which is undesirable when mounting fitted details or when fitting shaft journals in sliding bearings.

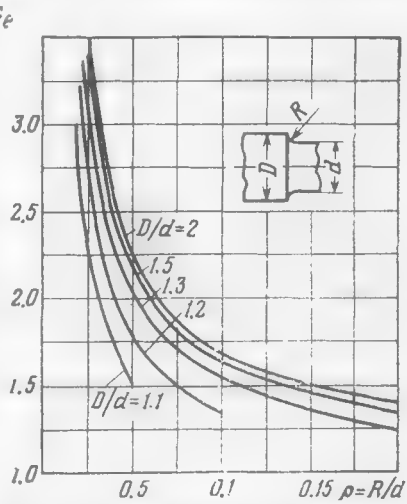


Fig. 217. Effective stress concentration coefficient of a stepped shaft in flexure as a function of relative fillet radius ρ = *R*/*d* and ratio *D*/*d* of adjacent diameters (after Serensen)

The reduction in the length of a shaft's cylindrical portion can be avoided by using face undercut fillets (Fig. 216*g*).

Face undercut fillets in terms of their effectiveness approximate round fillets, having similar *R*/*d* values.

Face undercutting is recommended when cylindrical shafts are mated with prismatic components, when there is place for a fillet of sufficiently large radius.

Figure 218 shows how higher strength fillets can be overlapped when mounting fitting components, for example, antifriction bearings, having small lead radii or small chamfers. For large-radius and elliptic fillets (Fig. 218*b*, *c*), intermediate washers which have appropriate recesses are used.

Fillets must be provided at all transitional entering angles of parts subjected to high cyclic loads (Figs 219 and 220).

Holes. Stress concentrations caused by holes can be reduced by enlarging the sections of a part where holes are located by rounding

hole rims, crimping of edges, strain-hardening hole walls and caulking the material around the periphery.

Figure 221 illustrates the sequence of operations when machining holes in highly loaded components (e.g., discharge holes of turbine

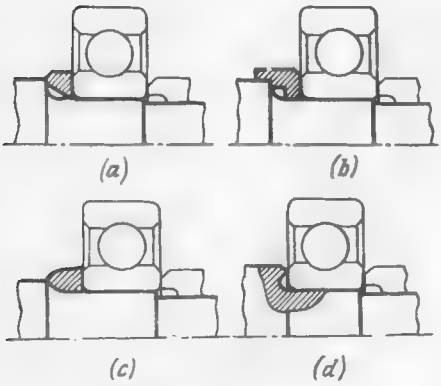


Fig. 218. Installation of ball bearings on shafts with strengthened fillets

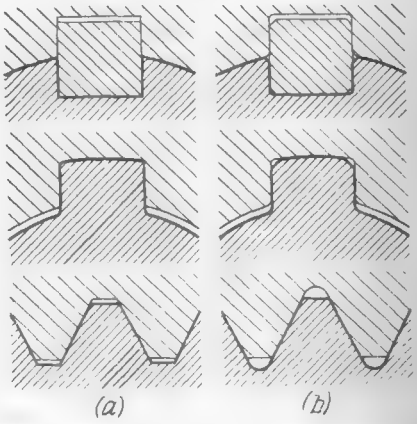


Fig. 219. Shapes of some typical machine elements
(*a*) irrational; (*b*) rational

disks): *a*—drilling, *b*—chamfering, *c*—countersinking, *d*—reaming, *e*—rounding of hole rims, *f*—compacting the fillet, *g*—broaching the hole with a ball.

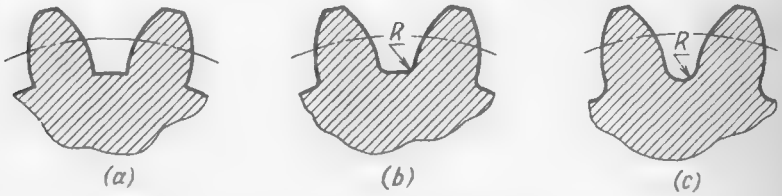


Fig. 220. Conjugation of gear teeth with rim
(*a*) irrational; (*b*) improved; (*c*) rational

Hollow shafts. The internal cavities of critical hollow shafts undergoing high cyclic loads, should be machined to the highest, economical surface finish: grinding, polishing, rolling, sizing, compacting broaching, etc. Recesses, threads and other stress concentrators should be avoided on inner surfaces.

In stepped holes smooth transitions between steps should be introduced. Incorrect designs are presented in Fig. 222*a*, *b*. The acute

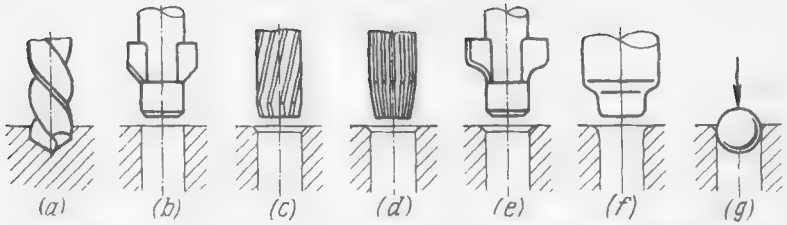


Fig. 221. Successive stages in machining holes in cyclically loaded parts

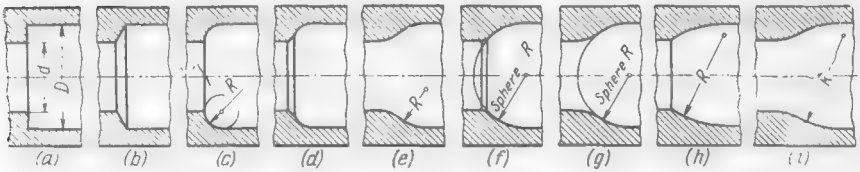


Fig. 222. Shapes of stepped holes in hollow shafts

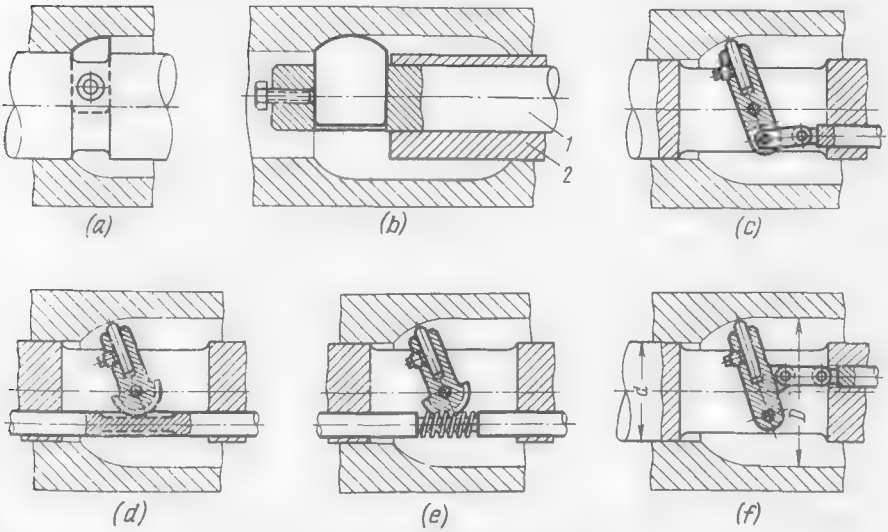


Fig. 223. Machining internal fillets

entering angles close to steps cause stress concentrations and impair shaft strength. The versions presented in Fig. 222*c*, *d* show how fillets improve strength. Fig. 222*e-i*, illustrate shafts with bottle-neck-shaped holes.

Fillets in bottle-neck-like holes are turned to a templet, which monitors the cross-slide traverse. Finish turning is effected by a form tool, secured in a boring bar, centered in the smaller diameter hole (Fig. 223*a*).

More difficult to machine are hollows with limiting fillets at both ends (barrel-like holes). One machining technique is illustrated in Fig. 223*b*. The form tool is clamped in rod *1*, secured eccentrically in boring bar *2*. By turning the rod, the tool is retracted, the boring bar is introduced into the hole and the cutting tool then moved into position.

The most productive method of machining bottle-necked holes is with a swing tool mounted in a boring bar and monitored with a connecting rod (Fig. 223*c*), rack (Fig. 223*d*) or worm gearing (Fig. 223*e*).

The fillet radius is determined by the position of tool-holder.

The most advantageous swing mechanism design is when the tool-holder axis is positioned at the boring bar centre (Fig. 223*c*, *d*), i.e., when the fillet forms a sphere with its centre on the shaft axis (see Fig. 222*f*, *g*). Such a shape ensures a smooth transition from one hole diameter to another.

An even smoother transition is obtained by shifting the tool-holder pivot point off the shaft axis (Fig. 223*f*).

To determine the maximum radius of an internal fillet the following approximate formula can be used

$$R_{max} = 0.5 (D + 0.7d)$$

where D and d are the maximum and minimum hole diameters, respectively (see Fig. 222*a*).

When machining inner steps in preformed large tube blanks the material fibres are cut through at the most stressed areas of transition from one step to another.

To attain better strength, shafts with barrel-shaped interiors have their tube ends hot pressed (Fig. 224). Used as blanks are thick-wall drawn tubes whose outside diameter is turned with extra metal for reduction of ends (Fig. 224*a*). Surface *m* is the datum for subsequent operations.

The ends are reduced by pressing (Fig. 224*b*) until the hole is fully closed (Fig. 224*c*). Then, from surface *m* holes *n* are bored for shaft journals and the barrel-shaped bore *h* machined employing one of the above-listed techniques (Fig. 224*d*).

Then using holes *n* as a datum, the outside diameter is finish machined.

Crankshafts. An example of successive fatigue strength enhancement of crankshaft throws is pictured in Fig. 225.

The initial design (Fig. 225*a*) has low strength. In the version, illustrated in Fig. 225*b*, the strength is increased due to the larger

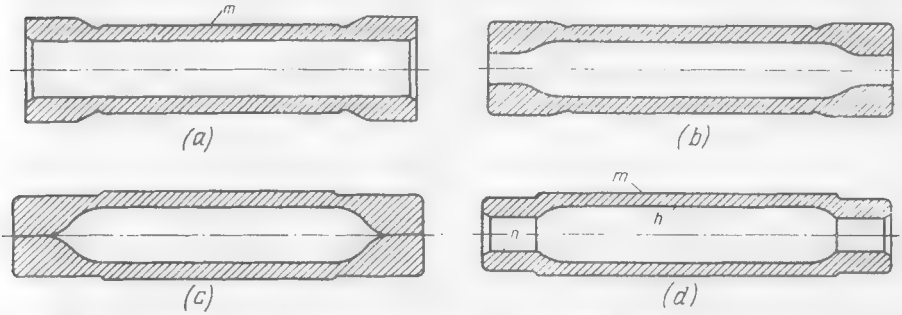


Fig. 224. Process stages im manufacturing one-piece forged shafts with barrel-shaped interior

diameters of the main and rod necks and increased web sections. The increased neck diameters shorten lengths m between necks parts which are most critical in terms of strength.

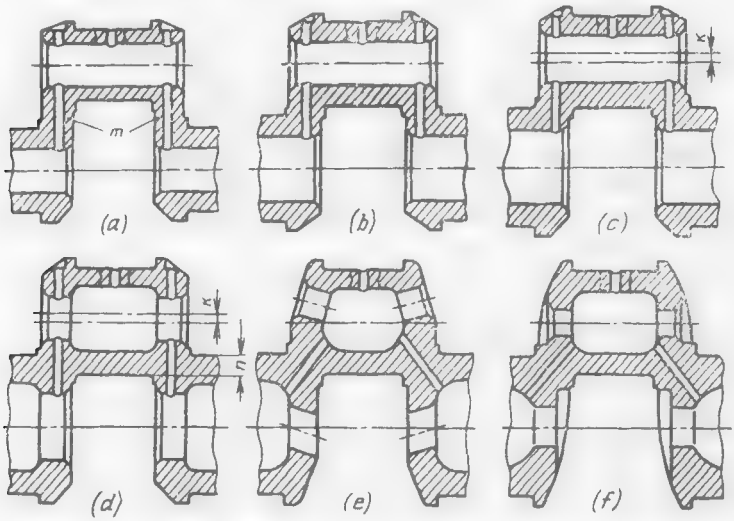


Fig. 225. Enhancing the fatigue strength of a crankshaft

Still better strength is obtainable by offsetting the conrod neck internal recess from the neck geometric axis by value k (Fig. 225*c*): this will strengthen the connection between rod necks and webs and increase neck strength resistance against ignition flexure.

In the most rational design (Fig. 225*d*) the neck diameters are increased until the main and conrod necks overlap, thereby ensuring direct connection of necks (part *n*). A barrel-shaped bore between the main and conrod necks lowers stress concentration from oil holes in crankshaft webs and betters the neck-to-web connection strength. A combination of all of the above-described measures substantially improves crankshaft strength.

Forms with deep internal cavities are practicable in cast crankshaft designs (Fig. 225*e*, *f*).

(*b*) *Elimination of Load Concentrations*

A most important design rule for cyclically loaded parts is the elimination of local stress surges arising where the concentrated loads are applied.

Let us take, for example, the design of gear teeth (Fig. 226*a*). Out-of-straightness of spur gear teeth, helix errors of helical gear

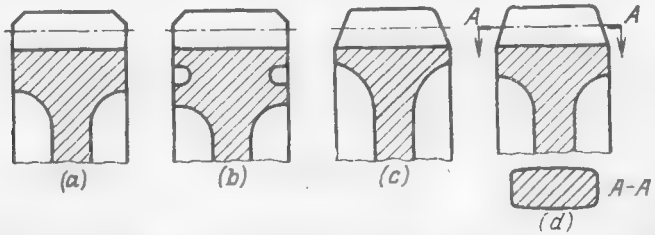


Fig. 226. Elimination of load concentration on tooth faces

teeth, misalignment of gear axes as a result of improper mounting or inaccurately spaced supports—these and/or other defects may cause load concentrations on tooth end faces, the consequence being higher bending and crushing stresses.

It is advisable to increase tooth compliance at the end faces by the introduction of relieving recesses in the gear rims (Fig. 226*b*) or by lessening rim rigidity near the end faces (Fig. 226*c*).

An effective means of preventing high-edge pressures is by giving teeth a barrel-shaped form (camber) simultaneously rounding-off face edges (Fig. 226*d*). This method is beneficial because even in the event of distortions and inaccuracies the gear contact spot will remain approximately in the tooth centre, thereby assuring most favourable loading of teeth.

Pressed connections. Presented in Fig. 227 are some ways of strengthening pressed connections. The simplest of these consists in increasing the diameter of fitting surface (at least by 5-10%) in relation to the shaft main diameter *d* (Fig. 227*a*).

Advisably lowered are the stresses at the edges of pressed fitted connections by relieving recesses in the hub (Fig. 227*b*), shaft

(Fig. 227*c*), by tapering the hub down towards its end faces (Fig. 227*d*) or by cambering the shafts (Fig. 227*e*).

Shafts strength can be improved by rolling load relieving grooves near the joint ends (Fig. 227*f*).

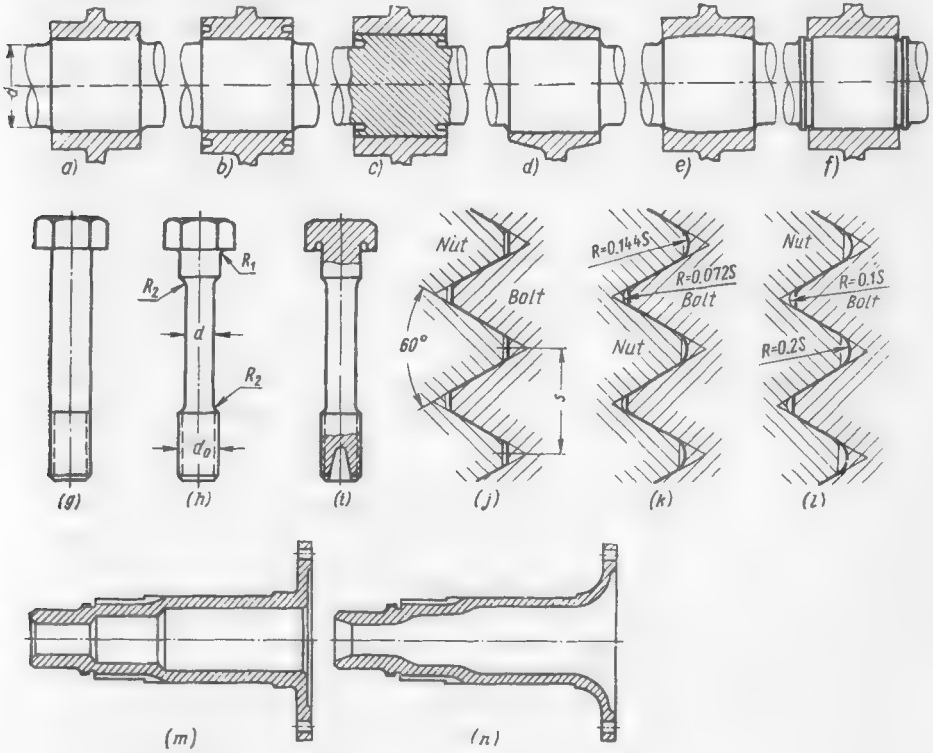


Fig. 227. Improvement of fatigue strength

The most effective method of improving fatigue strength of pressed connections is by strengthening the contacting surfaces through thermal-chemical treatment and plastic deformations (rolling).

Fasteners. Constructional ways of increasing the fatigue strength of fasteners (bolts, studs, etc.) largely means increasing their elasticity, thus assuring a lowering of maximum stresses and stress amplitudes. If bolt lengths are the same higher elasticity can be achieved by lessening the bolt stem diameter d to 0.7-0.8 of its thread nominal diameter d_0 (Fig. 227*h*); in this case the sections of the bolt stem and thread become approximately equistrong.

The head-to-stem bolt transition must have the maximum possible radius or face undercuts (Fig. 227*i*). Equally necessary are smooth fillets (R_2) at the stem-to-thread transit section.

The strength of thread itself is strongly dependent on fillet radii near the bolt thread root (this effect is less in a nut because it is usually stronger than a bolt).

The current standard (GOST 9150-59) does not stipulate the root-form between bolt threads: they may be flat (Fig. 227*j*) or rounded to a radius (Fig. 227*k*). In this case the maximum radius for bolts is equal to $0.144s$ (s = thread pitch).

For critical screwed connections subjected to high cyclic loads it is advisable to employ threads with more stream line-rounded root radii: for nuts $R = 0.1s$, for bolts $R = 0.2s$ (Fig. 227*l*).

Particularly high fatigue strength is obtained by rolling rough-cut and heat-treated threads and also when rolling threads outright on the solid metal.

As a general rule parts subjected to high cyclic loads should have smooth forms to assure evenness of force flow. To avoid stress jumps in the sections of parts it is necessary to determine from conditions of approximate similarity the stresses by considering all active loads.

Well designed parts undergoing high cyclic loads have a characteristic smooth form (Fig. 227*n*), often conveniently called "streamline".

5.3. Cylindrical Joints Operating under Alternating Loads

Any joint transmitting a pulsating torque or subjected to alternating radial loads undergoes work hardening, welding and fretting corrosion.

In the main these defects are caused by numerous repetitive strains and microshears in the mated surfaces, in the circumferential and lengthwise directions, which heat the material. Joints, operating in the severest conditions, are sometimes heated to 500-600°C. Momentary peak temperatures at parts of micro-irregularity contiguities reach 1000-1500°C.

Under such circumstances work hardening easily occurs, crushing the surface, forming irregularities and partial cohesion of mated metallic surfaces. During the next stage such surfaces may weld together so that their disengagement is possible through destruction only.

This kind of welding can occur at a temperature well below the welding temperature of the material. Under usual conditions the metal surface becomes coated with strong adsorbed films of lubricants, oxides, moisture and vapours, which prevent direct metal-to-metal contact. Heat and excessive pressures, particularly when at points of conjugated microirregularities destroy the films, and the metal particles close to a distance, at which the molecular and crystalline interaction forces cause metal fusion.

Greater propensity to welding is shown by identical metals and metals which have similar crystalline structures. Structural heterogeneity presence of several phase changes and non-metallic inclusions (carbides, oxides) in metals stop the welding phenomenon. Ample resistance to welding is offered by hardened steels with a martensite structure (provided, however, no tempering has occurred due to overheating).

Frictional (fretting) corrosion is, in fact, oxidation of surfaces of metals as a result of microshear caused by local elevations of temperature. On steel and cast iron surfaces iron oxides form (mostly Fe_2O_3) either as rusty spots or (if the process has developed far enough) as clusters of brown powder. On bronze surfaces appear green films of copper oxides.

Work-hardening and welding defects can be averted by:

reducing strains and microshear in mated surfaces (making the design more rigid, employing power-clamped joints, clearance-free torque transmissions);

removing heat, evolved during micro-displacements (applying gaskets made from heat-conducting materials, use of cooling oil in joints having clearances);

application of separating coatings (phosphating, copper-cladding, tinplating, cadmium-plating, galvanizing, coating with polymeric films, introduction of solid lubricants having a molybdenum disulphide base, colloidal graphite, etc.);

increasing hardness and thermal stability of the surface layer; creating structures which show good resistance to welding (aluminizing, sulphiding, nitriding, diffusion chrome-plating, boronizing (see Table 24).

The main design method, enabling work-hardening and the welding to be avoided, is to provide interference fits to the mating surfaces, radial (on cylindrical surfaces) or axial (on end faces). An interference fit materially enhances the rigidity of a unit as a whole, lessens elastic deformations in a system and efficiently constrains relative displacements of mated surfaces.

Figure 228 shows some methods of fitting parts to shafts.

Mounting without tightening or with a weak clamping (Fig. 228*a*, *b*) is unacceptable for power-transmitting connections.

Often used now is the axial tightening of a hub resting against a shaft shoulder (Fig. 228*c*). Here the amount of radial interference depends upon the grade of the hub fit. The heavier the operating conditions, the tighter should be the fit. In shaft-end connections a central bolt is also used for clamping (Fig. 228*d*), or an internal nut (Fig. 228*e*) which provides for a greater tightening.

A purely radial interference is provided by a press fit (Fig. 228*f*). By introducing taper pins (Fig. 228*g*) into the joint a practically clearance-free torque transmission may be obtained, thus eliminating angular micro-displacements of the mating surfaces. It should be

Table 24

Measures Which Prevent Welding

Process	Essence of process	Procedure	Purpose
Phosphating	Deposition of a crystalline phosphide film on the surface	Treatment in a solution of phosphates Fe, Mn and Zn	Creation of a separate wear-resistant film
Aluminizing	Deposition of crystalline film Al_2O_3 on the treated surface. Formation in the surface layer of Al solid solutions in α-iron	Soaking in a blend of ferroaluminum powder and Al_2O_3 at 900-1000°C (5-8 h)	Increases thermal and corrosion resistance of the surface layer
Sulphiding	Formation of iron sulphides in the surface layer	Soaking in a blend of sulphates ($NaS \cdot 9H_2O$) and cyanates (catalyzers) at 550-580°C (2-4 h)	Imparts anti-seizure properties; increases welding resistance
Nitriding	Formation of Fe, Al, Mo hard nitrides etc. in the surface layer	Soaking in an ammonia atmosphere at 500-550°C (20-30 h)	Enhances thermal and corrosion resistance and also hardness (800-1200 VPH)
Diffusion chrome-plating	Formation in the surface layer of chrome carbides and Cr solid solutions in the α-iron	Soaking in a medium of volatile chrome chlorides $CrCl_2$, $CrCl_3$ (gaseous chrome-plating) at 800-1200°C (5-6 h)	Betters thermal resistance and hardness (1200-1400 VPH)
Boronizing	Formation in the surface layer of Fe borides and B solid solutions in the α-iron	Soaking in a blend of powders of boron carbide (B_4C) and borax ($Na_2B_4O_7$) at 900-1100°C (5-6 h)	Increases thermal resistance and surface hardness (1500-1800 VPH)

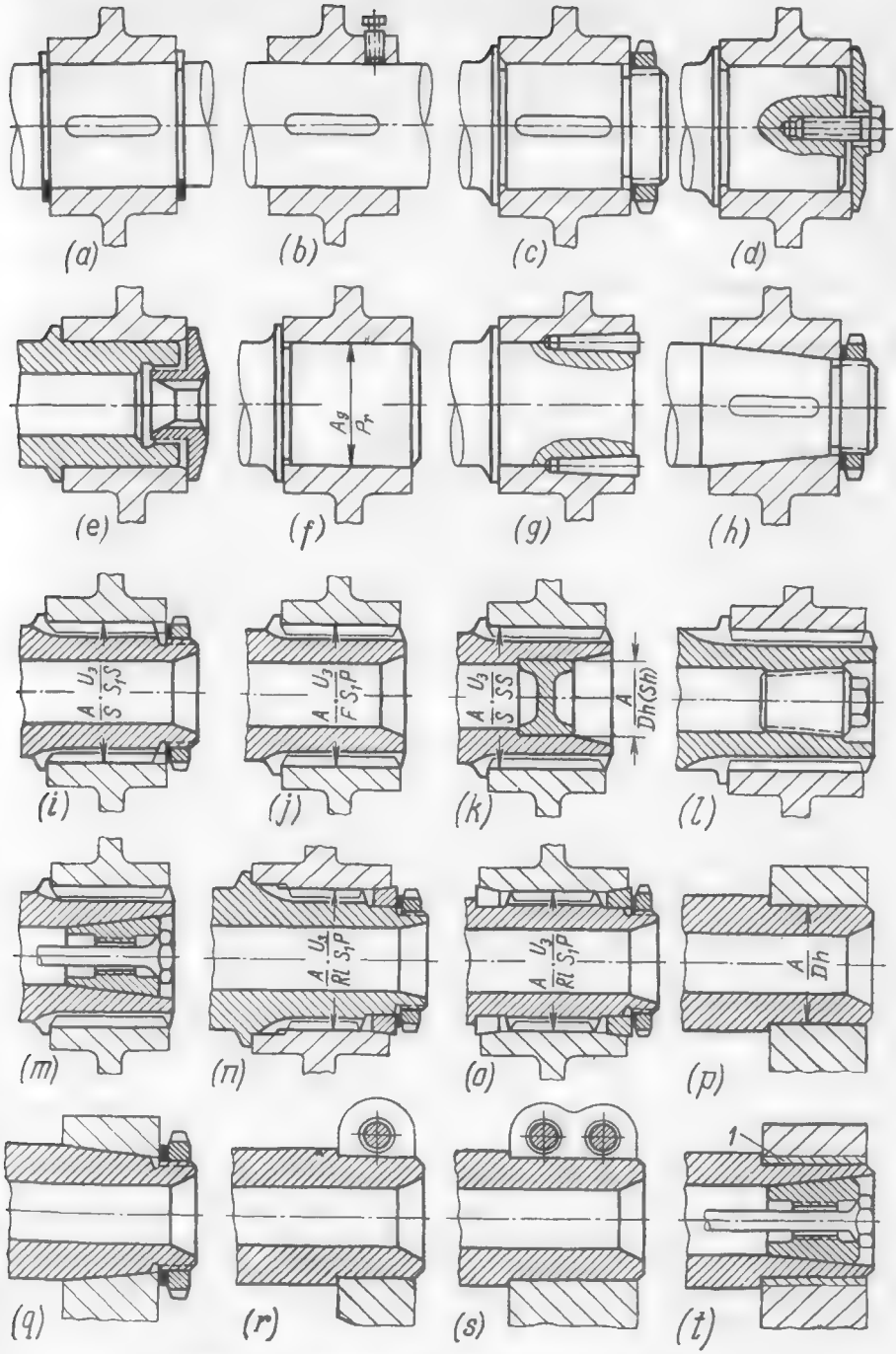


Fig. 228. Tightening of cylindrical connections

emphasized, however, that in this case a permanent joint is obtained.

A rather strong connection is attained when tightening on a taper (Fig. 228*h*). The value of radial interference will depend on the force, with which the nut is tightened. In particular instances it is obligatory to use torque wrenches although this method does not fully guarantee the degree of tightening since it depends to a great extent upon the state of the mating surfaces.

Slide-fit splined connections (*C* to the aligning diameter, *S* and *C* to the splines working faces) must be tightened by a nut (Fig. 228*i*).

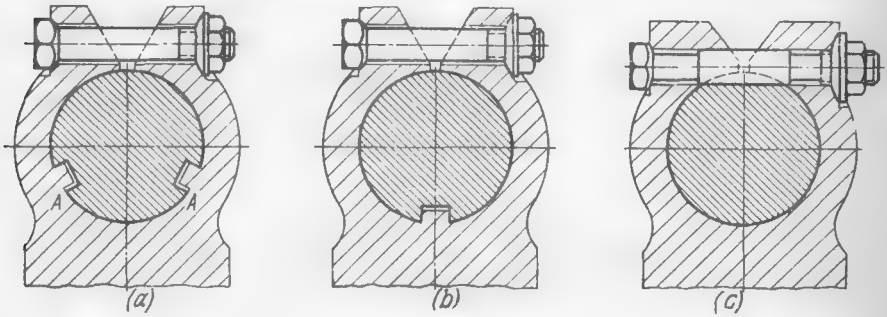


Fig. 229. Clamp connections

For permanent or rarely disconnected connections a press- or force-fit on the aligning diameter is usually practised or a drive-fit to the working faces (Fig. 228*j*).

Figure 228*k*, illustrates how radial interference can be obtained by press-fitting a plug into the shaft bore. A permanent connection is thus accomplished. If dismountable connections are required, the above tightening is implemented either by a taper threaded plug (Fig. 228*l*) or with a cone tightened by a central bolt (Fig. 228*m*). In the latter case the plug must be internally threaded to allow use of an extractor.

In critical heavy-duty splined connections to hubs are tightened on tapers (Fig. 228*n*, *o*).

When connecting cylindrical components with prismatic elements (e.g., journals to webs in split-type crankshafts) use is made, apart from the above-listed techniques (press- and taper-fits, Fig. 228*p*, *r*) of clamps (Fig. 228*s*, *t*), as well as internal conical plugs (Fig. 228*u*). To avoid work-hardening the mating surfaces, a thin-walled liner *1* made from bronze or sheet brass is used.

When dealing with clamp connections it is necessary to assure uniform tightening throughout the entire circumference of the clamp. Fig. 229*a*, illustrates an incorrect design. The torque from the journal

to the web is transmitted by two tenons. During tightening the upper edges of the tenons thrust against slot walls and the AA section remains loose hence work-hardening here is unavailable.

In a correct design (Fig. 229*b*) the tenon is positioned at the axis of clip symmetry. Uniform tightening is also obtained when the torque moment is transmitted by a bolt fitted into a groove machined in the journal.

Owing to the above clamps cannot be used for heavily loaded splined connections.

Contact strength

With contact loading the force acts upon a rather small area of surface, thereby causing higher stresses in the metal surface layer. This kind of loading is encountered most frequently when spherical and cylindrical bodies come into contact with flat, spherical or cylindrical surfaces. Such are, for instance, antifriction bearings, gears, rollers of overrunning clutches, friction speed variators, etc.

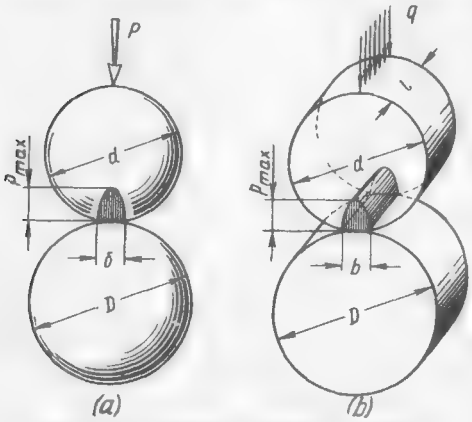


Fig. 230. Contact loading schemes
(*a*) spheres; (*b*) cylinders

In the theoretical solutions on the stressed state of elastic bodies (Hertz, Belyaev, Fapple) in the zone of contact it is assumed that the applied load is static, that the materials of the contacting bodies are homogeneous and isotropic, that the contact area is small in comparison with the total surface area and that the acting force is directed normally to the area of contact.

In the zone of contact a flat area is formed whose dimensions depend upon the elasticity of the materials and the shapes of the compressed bodies. During compression of spheres (Fig. 230*a*) this area will have the shape of a circle with diameter

$$\delta = 1.4 \sqrt[3]{\frac{P\vartheta}{E}} \text{ mm}$$

where P = load, kgf

$E = \frac{E_1 E_1}{E_1 + E_2}$ = reduced modulus of elasticity for the material of the spheres, kgf/mm²

$\vartheta = \frac{dD}{D \pm d}$ = reduced diameter of spheres, mm (the minus sign relates to the case when a convex surface contacts a concave surface with diameter D)

The maximum pressure in the centre area is 1.5 times greater than the average pressure

$$p_{max} = 1.5 \frac{P}{0.785\delta^2}$$

When cylinders are compressed (Fig. 230*b*), the area will have a reactangular form whose width

$$b = 1.5 \sqrt{q \frac{\vartheta}{E}} \text{ mm}$$

where ϑ = reduced diameter of cylinders
q = load upon cylinder unit length, kgf/mm²

The maximum pressure along the centre line of the area is 1.27 times higher than the average pressure

$$p_{max} = 1.27 \frac{q}{b}$$

The material fibres in the zone of acting maximum pressures are in a state of omnilateral compression; also in the fibres form mutually perpendicular compressive stresses σ_x, σ_y, σ_z and directed to them at 45° octahedral shearing stresses $\frac{\sigma_z - \sigma_y}{2}$, $\frac{\sigma_z - \sigma_x}{2}$, $\frac{\sigma_y - \sigma_x}{2}$. The distribution of these stresses (in fractions of maximum pressure p_{max} per contact area) in terms of surface layer depth (in fractions of contact area width b) is shown in Fig. 231. Normal stresses (Fig. 231*a*) have their greater value ($\sigma_z = \sigma_y = p_{max}$; $\sigma_x = 0.5 p_{max}$) on the surface, while shearing stresses (Fig. 231 *b*) are maximum at a distance of (0.25-0.4) b from the surface.

Under omnilateral compression the yield point of hardened high-strength steels reaches 300-500 kgf/mm², which is approximately 4-5 times that under unilateral compressive stresses.

In mechanical engineering structures the load, as a rule, is cyclic, due to the periodically varying active force, as well as the usual relative movements of contacting bodies.

Pictured in Fig. 232 are the main operating schemes of coupled bodies with contact load (in the descriptions design analogies are given in brackets):

a—static load (lifting screw with spherical end); *b*—impact load (tappet with spherical end); *c*—sphere rotating about axis normal to contact area (ball toe); *d*—sphere moving parallel to support surface (lever—mechanism with spherical striker); *e*—sphere rolling over support surface (ball-mounted straight-line guide); *f*—sphere rolling and simultaneously moving parallel to support surface (rolling accompanied by skidding); *g*—transmission of rotation from one cylinder to another without resistance on driven cylinder (rolling of a cylindrical surface by a roll); *h*—same but with resistance on driven

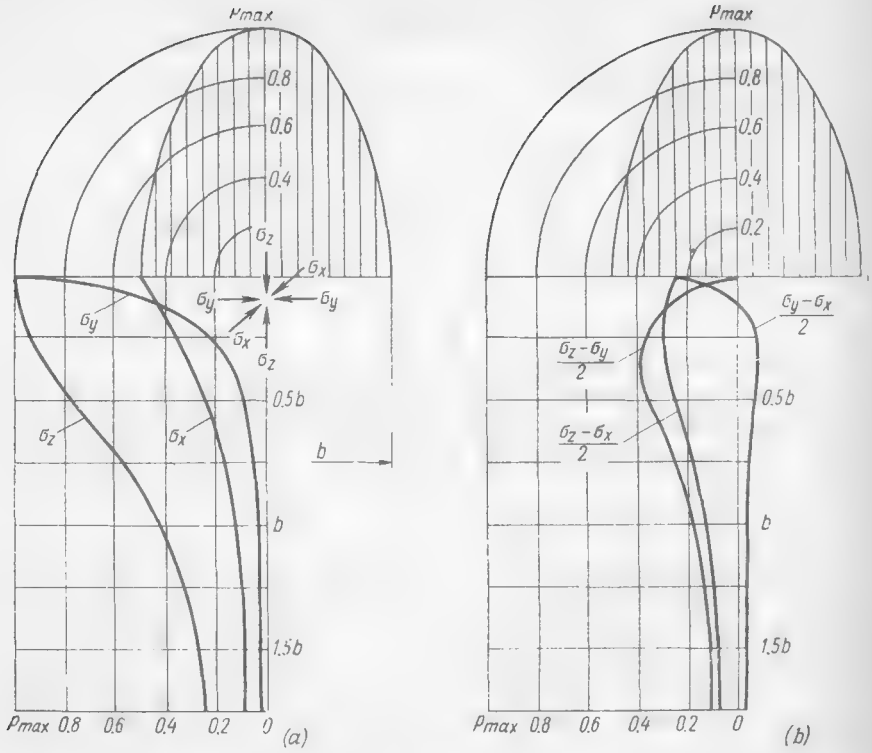


Fig. 231. Distribution of stresses near the surface layer in the contact area (compression of cylindrical bodies)

(a) normal stresses; (b) tangential stresses

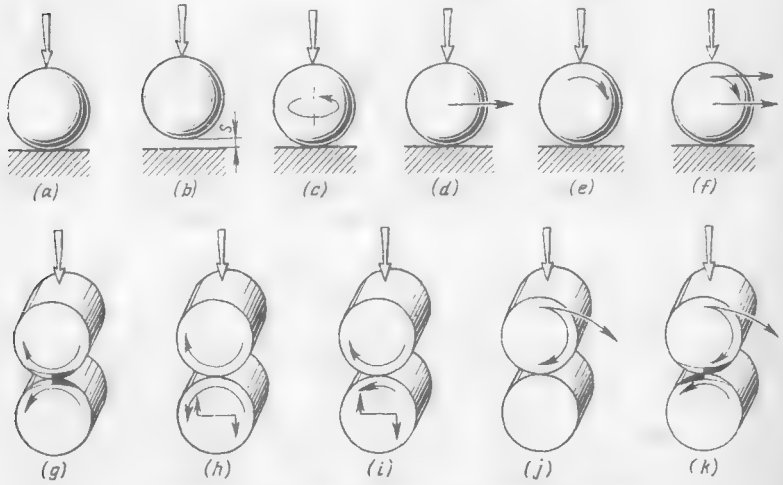


Fig. 232. Typical cases of contact loading

cylinder (disks of friction speed variator); *i*—same but accompanied by slipping; *j*—rolling of one cylinder over another (roller bearing); *k*—same but accompanied by slipping in-between cylinders (gear teeth).

In the schemes Fig. 232*e-k*, the load bears cyclic character even if the acting force is static; the load is exerted successively over various points on the surface.

Relative movements of contacting bodies disturbs Hertz's stress distributions in the zone of contact, where the surface layer is subjected to compression and tension in a tangential direction. Disposition of compressive and tensile zones depends on the kinematics of movement. In sliding (Fig. 232*d*) and pure rolling (Fig. 232*e*, *g*, *j*) the compression zones on both conjugated surfaces are found on one side of the contact centre (i.e., opposite to the movement), while on the other side the material is subjected to tension (Fig. 233*a*).

In rolling accompanied by slipping (Fig. 232*i*, *k*) the compression zone on the leading (anticipating) surface is in front of the contact centre (opposite to the movement), whereas on the lagging (delaying) surface it is behind; on the opposite lying areas the material is subjected to tension (Fig. 233*b*).

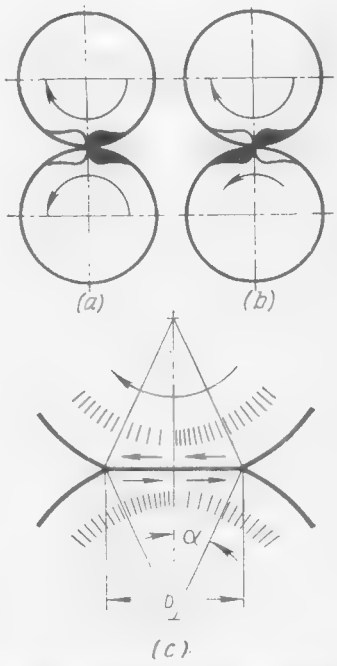


Fig. 233. Compression and tension in the zone of contact

In the leading surface compression zone (Fig. 233*c*) the fibres of material converge and shift in the direction, shown by the arrows. In the zone of tension the fibres are elastically stretched and shift in the same direction. On the lagging surface the fibres move in the opposite direction. As a result, friction forces appear on the contacting surface deflecting the acting forces from the normal to the area of contact. The tangential compression and tension change the type of the state of stress in the contact zone.

Periodic compression and tension of fibres causes, even in the case of pure rolling with resistance on the driven body (see Fig. 232*h*), systematic lagging of the driven body. The length of surface of the leading body at angle α (see Fig. 233*c*) is equal to $b/2 - \Delta b$, where

Δb—elastic shortening of the surface. The length of surface of a driven body over the same area will be $b/2 + \Delta b'$, where $\Delta b'$—elastic lengthening of the surface. Hence, the rotary speed of the driven body is less than that of the driving one by the ratio

$$i = \frac{b/2 - \Delta b}{b/2 + \Delta b'} = \frac{1 - 2\Delta b/b}{1 + 2\Delta b'/b}$$

In practice, $i = 0.99\text{-}0.995$.

Clearly from the above actual conditions in the zone of contact are more complex than those obtained under a static load, thus formulae deduced for a static load can only be applied as a first approximation.

With contact loading durability of cyclically loaded connections is determined from the fatigue strength of material. The fatigue strength curves are dependent on the kind of loading. In general they are similar to the fatigue curves, taken for the case of a uniaxial stressed state (tension, compression, bending), the difference being that the numerical values of breaking stresses are much higher and the curves do not have a clearly expressed plateau (fatigue limit).

The hardness of the surface layer is the fatigue strength decisive factor under contact load conditions (Fig. 234).

The process of fatigue failure in rolling, as well as with contact surfaces sliding at a low speed, develops rather peculiarly. Initial cracks emerge in the zone, where maximal tangential stresses are active—i.e., at a depth, equalling approximately to 0.3-0.4 of the contact area size. While gradually developing, the cracks propagate onto the surface, producing a typical pit-like eruption pattern. Further on these pits grow in size and blend into chains, in-between which large metal particles flake and spall off. This phenomenon is called pitting, which, as a rule, leads to failure as the pits unite together.

Increasing speeds of the relative movements (rolling accompanied by slipping) have up to a limited degree a beneficial effect. In the process of wear the damaged layer is gradually removed and, consequently, no spalling off occurs. The durability of a joint will now be dependent upon the intensity of the abrasive wear, changing in the course of time the original shape of the contacting surfaces.

A typical example of a contact fatigue failure is the pitting of gear teeth flanks. Here the pitting is generally concentrated on the areas of the teeth lying most closely to the gear pitch circle. This can be attributed to the fact that at conventional values of the overlapping coefficient $\varepsilon = 1.2\text{-}1.8$ the loading upon these areas is taken up by one tooth only, whereas the loading, exerted onto the areas, close to the heel and toe, is shaped by two teeth. Moreover, while middle area of tooth profile roll without slip, the portions close to the heel and toe roll with slip. Thus, the latter areas are, in effect,

being ground by conjugated surfaces, eliminating surface damage, but in time leading to distortion of the involute profile.

A lubricant acts in two ways. Under moderate pressures an oil film in the zone of contact contributes to a more uniform pressure distribution and increases the actual contacting surfaces. The sur-

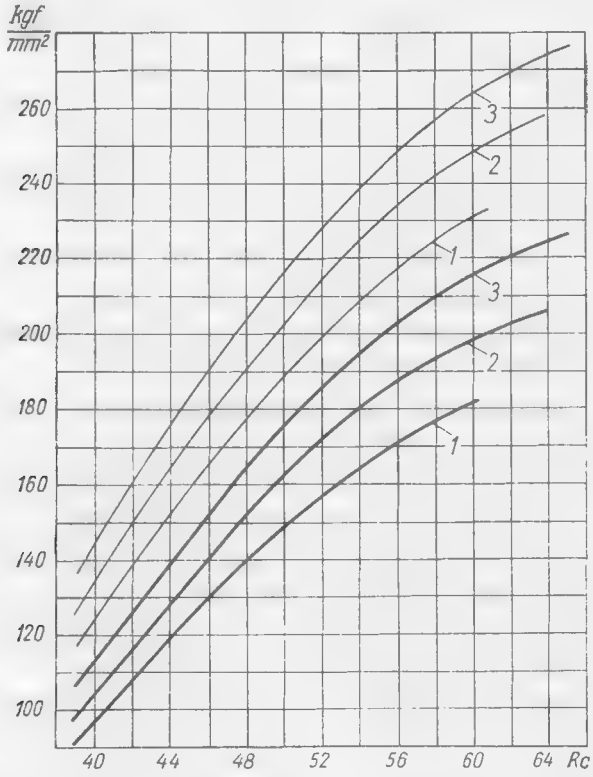


Fig. 234. Fatigue contact strength as a function of Rockwell hardness *C*

1 — steel grade 45ХН (induction hardened); *2* — steel grade 20Х2Н4А (case hardened); *3* — steel grade ШХ15 (hardened and low tempered). Heavy lines — ultimate stresses at $N = 10^7$ cycles; fine lines — at $N = 10^8$ cycles

face rolling produces a certain hydrodynamic effect: higher pressures occur in the film, forced from the gap which helps separation of the metallic surfaces.

The hydrodynamic effect is still more expressed when slip is present: the oil, entrained by the moving surface, is continuously fed into the narrowing part of the gap, thus separating the metallic surfaces. Under favourable conditions (high slipping speeds, small unit pressures, higher viscosity of oil) fluid-type friction begins.

Under high pressures the oil in the zone of contact has a negative influence. Under the influence of the running surface and capillary attraction, oil will penetrate into loose spots and microcracks expanding them, causing accelerated metal spalling. This phenomenon is particularly sharp when one of the surfaces in the high pressure zone is subjected to tension (Fig. 233*b*) this helping microcracks to open.

The problem of increasing the strength of contacting joints consists, first of all, in reducing contact area pressures by giving the conjugated surfaces as rational a form as possible.

6.1. Spherical Joints

The maximum stress σ_{max} in a surface layer, during compression of two spheres according to Hertz

$$\sigma_{max} = 0.6 \sqrt[3]{\frac{PE^2}{d^2}} \sqrt[3]{\left(1 \pm \frac{1}{a}\right)^2} \text{ kgf/mm}^2 \qquad (6.1)$$

where P = load, acting upon the joint, kgf

E = Young's Modulus for the material of the spheres, kgf/mm^2

d = diameter of the smaller sphere, mm

$a = \frac{D}{d}$ = ratio of diameters of the large and small spheres

The minus sign is for the case when the sphere acts on a concave spherical surface.

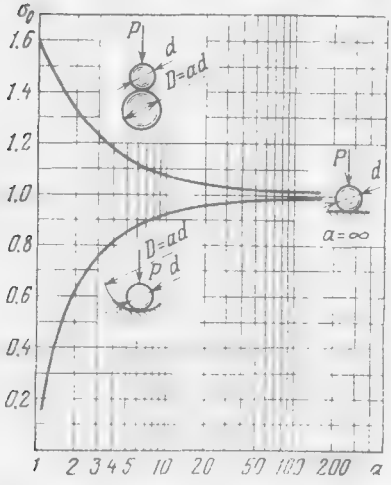


Fig. 235. Effective maximum stress σ_0 as a function of $a = D/d$ (compression of spheres)

With given d, P and E values the maximum stress will be proportional to the dimensionless value

$$\sigma_0 = \sqrt[3]{\left(1 \pm \frac{1}{a}\right)^2} \qquad (6.2)$$

The value can be defined as the maximum effective contact stress. The maximum actual stress is equal to the product of σ_0 and the factor $0.6 \sqrt[3]{\frac{PE^2}{d^2}}$.

The values of σ_0 as a function of a are shown in Fig. 235 for three kinds of loads: sphere-on-sphere, sphere on a spherical seating and sphere on a flat surface ($D = \infty$, $\sigma_0 = 1$). From the graph the following conclusions can be drawn.

The stress has its maximum value ($\sigma_0 = 1.59$) when two spheres of the same diameter ($a = 1$) are compressed. As the diameter of one

of the spheres increases, the stress falls becoming equal to $\sigma_0 = 1$ when $a = \infty$ (i.e., when a sphere rests upon a flat surface).

If a sphere is in contact with a concave spherical surface, the stresses will be significantly less than in the preceding case and will

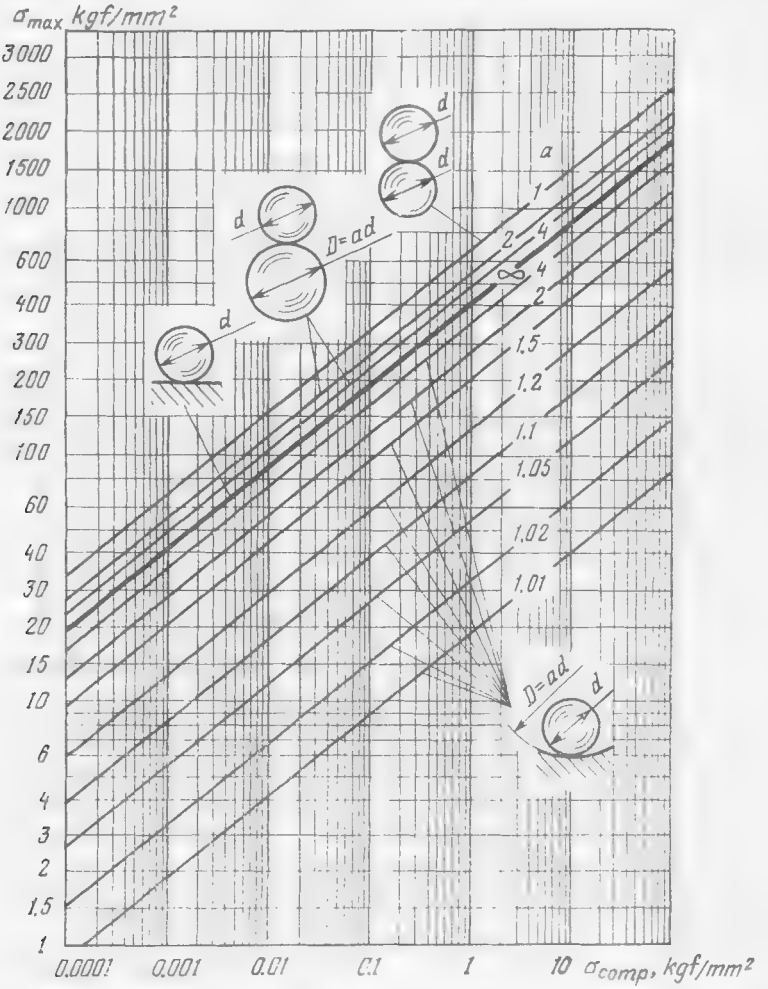


Fig. 236. Maximum stresses σ_{max} as function of compressive stresses σ_{comp} for different values of $a = D/d$ (compression of spheres)

sharply drop with the decrease in a, i.e., when the diameter of the concave spherical surface approaches closer to the sphere diameter; tending to zero at $a = 1$, when the concave spherical diameter is equal to the sphere diameter.

This does not imply that the stresses vanish completely; but only means that Hertz's formula is inapplicable when $a \approx 1$, because this renders invalid one of the assumptions underlying the basic theory, namely, the assumption that the size of the compression area is infinitesimal in contrast to the dimensions of spheres. When $a = 1$ (and even with values very close to unity) the stresses should be defined as crushing stresses.

Now, let us transform Eq. (6.1) as follows. Substitute the radicand $\frac{P}{d^2}$ by $\sigma_{compr} = \frac{P}{0.785d^2}$, where σ_{compr} is the compressive stress kgf/mm², produced by the action of force P in the central section of a d-diameter sphere (actual stress for solid spheres and conditional for truncated spheres and bodies having a limited spherical surface).

In the overwhelming majority of cases contact joints are made of steel ($E = 21\,000$ kgf/mm²). Substituting this value into Eq. (6.1) gives

$$\sigma_{max} = 430 \sqrt[3]{\sigma_{compr}\,\sigma_0} \tag{6.3}$$

where σ_0—effective stress Eq. (6.2).

The diagram, presented in Fig. 236, embraces the three kinds of loadings and shows σ_{max} as a function of σ_{compr} for various magnitudes of a and enables any problem connected with calculations of spherical joints to be easily solved.

A few examples for illustration.

Example 1. Find the maximum stress in a ball, 10 mm in diameter, resting upon a flat surface and loaded with force $P = 15$ kgf.

The compressive stress $\sigma_{compr} = \frac{P}{0.785d^2} = 1.27\,\frac{16}{100} \approx 0.2$. Starting from the point $\sigma_{compr} = 0.2$ on the abscissa moving vertically until meeting line $a = \infty$, we find on the ordinate axis that $\sigma_{max} = 250$ kgf/mm².

Example 2. The diameter of a ball and its load are the same. The ball rests in a spherical socket with $a = 1.02$. Clearly, the compressive stress in the ball is equal to that in the preceding case ($\sigma_{compr} = 0.2$). Now, moving upward from this point until meeting line $a = 1.02$, we find that $\sigma_{max} = 18$ kgf/mm².

Example 3. The given load is 100 kgf; permissible stress $\sigma_{max} = 50$ kgf/mm². Find the diameter of a ball suitable for these conditions when the ball is to be mounted in a socket with ratio $a = 1.02$.

Follow from the point $\sigma_{max} = 50$ kgf/mm² along the horizontal, until meeting line $a = 1.02$, we find on the abscissa that $\sigma_{compr} = 4$ kgf/mm². Hence, the ball diameter

$$d = \sqrt{\frac{1.27P}{\sigma_{compr}}} = \sqrt{\frac{1.27 \cdot 100}{4}} = 5.7 \text{ mm}$$

As the material in the contact area is subjected to omnilateral compression, high stresses (100-200 kgf/mm²) are generally permissible when designing contact joints. With shock loads the permitted stresses are lowered 2-3 times.

Contact joint members are made of thermally hardened steels, mostly of bearing grades ШХ15 and ШХ15СГ (hardness after hardening and low tempering 62-65 Rc).

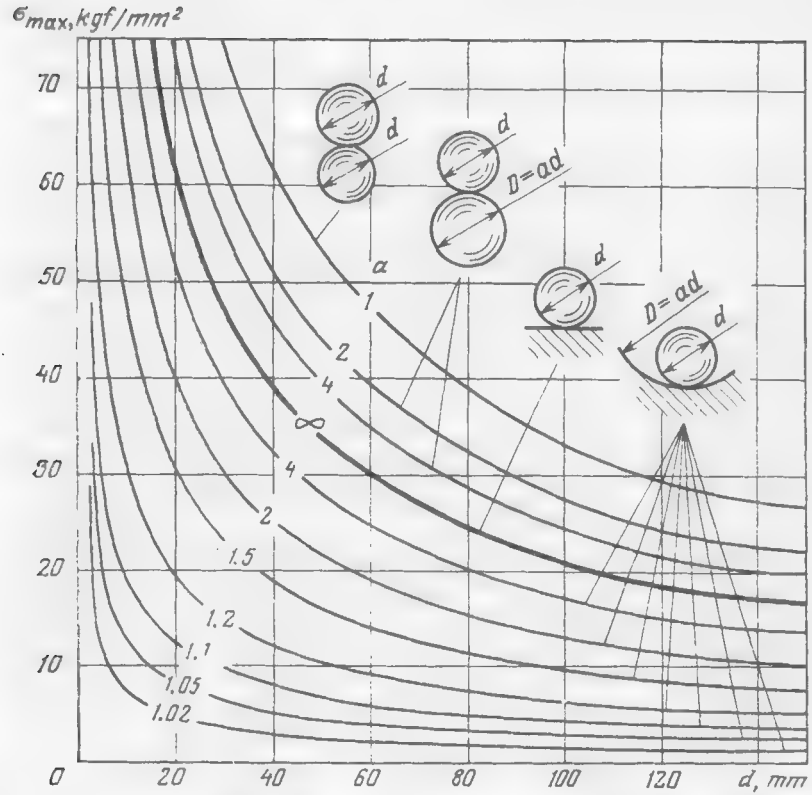


Fig. 237. Maximum stresses σ_{max} (with $P = 1$ kgf) as a function of sphere diameter d

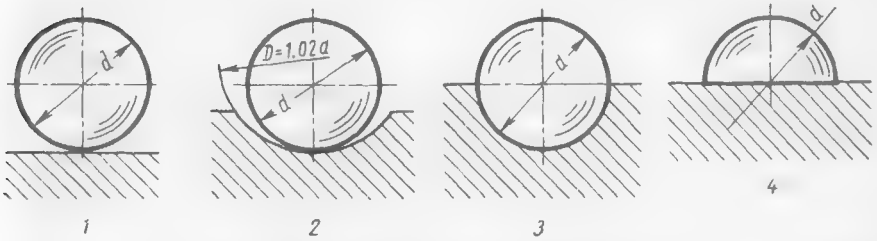


Fig. 238. Spherical connections

1 and *2* — point contact; *3* and *4* — surface contact

We now explain in a generalized form the influence of sphere diameter on strength. From Eq. (6.3)

$$\sigma_{max} = 430 \sqrt[3]{\sigma_{compr}}\,\sigma_0 = 430 \sqrt[3]{1.27 \frac{P}{d^2}}\,\sigma_0 \qquad (6.4)$$

Consequently, the maximum stress is inversely proportional to $d^{2/3}$.

The graph, plotted in Fig. 237, is based on Eq. (6.4). It is evident from the graph, that increase in sphere diameter sharply lowers stresses in the area of small diameters but as diameter increases the stress lowering proceeds more slowly. Beginning from a certain value of d, the lowering becomes hardly perceptible. In the given case when $a = 1.02\text{-}1.1$, this begins from sphere diameters of 20-30 mm. The range of advisable diameter increase is rather larger for spheres working on spheres.

In conclusion, let us compare the strength under point and surface contact (Fig. 238). Assume $\sigma_{compr} = 0.1$ kgf/mm^2 for all cases. For cases *3* and *4* having surface contact this stress, obviously, is equal to the bearing stress σ_b on the bearing surfaces ($\sigma_b = \sigma_{compr} = 0.1$ kgf/mm^2).

Listed below are the values of σ_{max}, for cases *1* and *2*, as well as the σ_{max}/σ_b ratios, characterizing the comparative strength of joints having point and surface contact. Since the permissible stresses under contact loading are, on average, 5 times higher than the admissible bearing stresses σ_b, the comparison is given in terms of the ratio $\sigma_{max}/5\sigma_b$

	Case 1	*Case 2*
σ_{max}, kgf/mm^2 . .	200	15
σ_{max}/σ_b	2000	150
$\sigma_{max}/5\sigma_b$	400	30

Obviously, the strength of a joint, when a sphere rests upon a flat surface, is 400 times, and when seating in a spherical socket at $a = 1.02$, is 30 times less than the strength obtained under surface contact.

6.2. Cylindrical Connections

According to Hertz, the maximal stress in cylindrical connections

$$\sigma_{max} = 0.6 \sqrt{\frac{PE}{ld}} \sqrt{1 \pm \frac{1}{a}} \text{ kgf/mm}^2 \qquad (6.5)$$

where P = acting force, on the connection, kgf
E = Young's modulus for the material of the cylinders, kgf/mm^2
d = diameter of small cylinder, mm
l = length of cylinders, mm
$a = D/d$ = ratio of large and small cylinders diameters

The minus sign is for the case when a cylinder acts on a concave cylindrical surface.

Denoting

$$\sqrt{\left(1 \pm \frac{1}{a}\right)} = \sigma_0; \quad \frac{P}{ld} = \sigma_{compr}$$

and assuming $E = 21\,000$ kgf/mm^2, we find that

$$\sigma_{max} = 87 \sqrt{\sigma_{compr}}\ \sigma_0 \text{ kgf/mm}^2 \tag{6.6}$$

where σ_0 = maximum effective contact stress

σ_{compr} = compressive stress, taken across the meridional section of a cylinder having diameter d

The values of σ_0 as a function of a are plotted in Fig. 239, for the three kinds of loads: cylinder on a cylinder, cylinder in a cylindrical seat and cylinder on a flat surface ($D = \infty$; $\sigma_0 = 1$).

In their general appearance the σ_0 curves are similar to the curves of the spherical connections (see Fig. 235). The stresses have their maximum value during compression of identical cylinders ($\sigma_0 = 1.41$), are lowered when a cylinder contacts a flat surface ($\sigma_0 = 1$) and sharply fall when a cylinder seats in a cylindrical socket, approaching zero if the diameter of the socket becomes equal to that of the cylinder ($a = 1$).

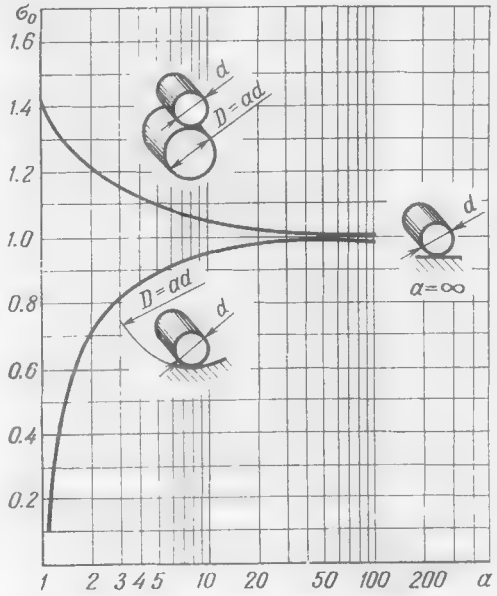


Fig. 239. Effective maximum stress σ_0 as a function of $a = D/d$ (compression of cylinders)

A diagram, used when designing cylindrical connections and based on Eq. (6.6), is presented in Fig. 240. While comparing this graph with the graph, shown in Fig. 236, one can clearly see, that the σ_{max} stresses, for the same σ_{compr}, values are much less in the case of cylindrical joints than for the spheres.

The ratio between stresses $\sigma_{max\ sph}$ in spherical connections and stresses $\sigma_{max\ cyl}$ in cylindrical connections agreeing to equations (6.3) and (6.6) is equal to

$$\frac{\sigma_{max\ sph}}{\sigma_{max\ cyl}} = \frac{430 \sqrt[3]{\sigma_{compr}}\ \sigma_{0\ sph}}{87 \sqrt{\sigma_{compr}}\ \sigma_{0\ cyl}}$$

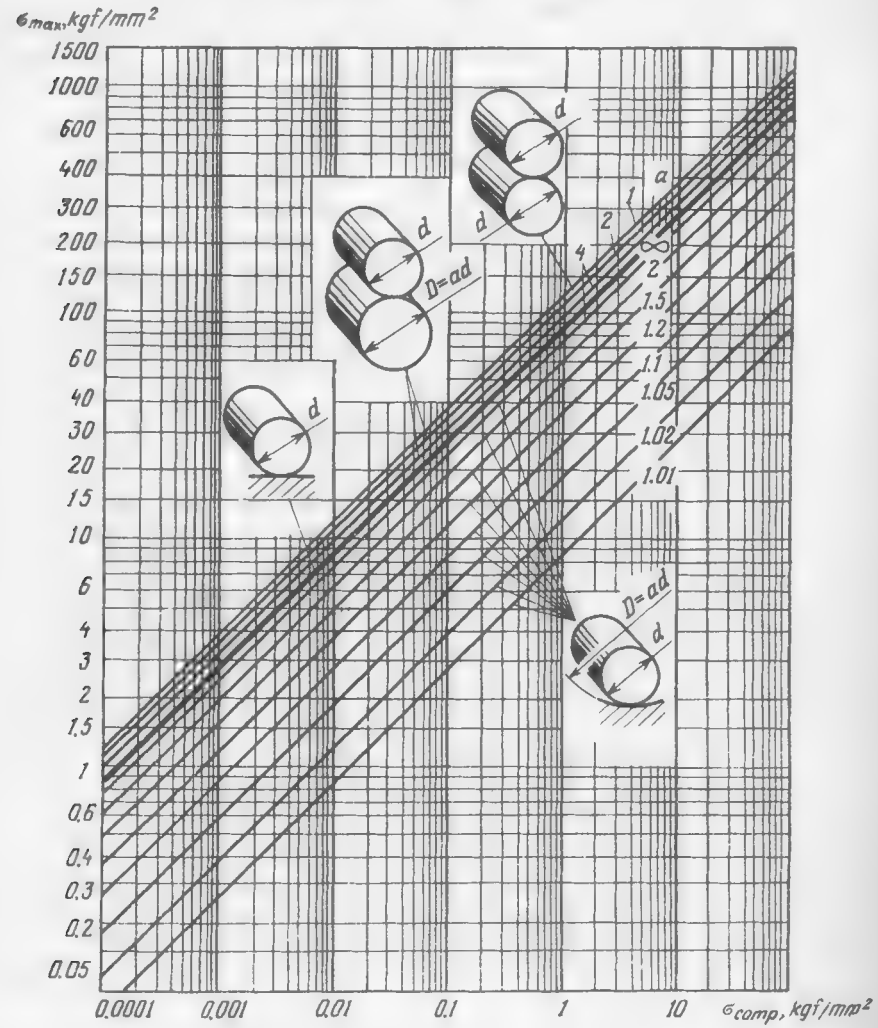


Fig. 240. Maximum stresses σ_{max} as a function of compressive stresses σ_{comp} for different values of $a = D/d$

If σ_{compr} is the same then

$$\frac{\sigma_{max\ sph}}{\sigma_{max\ cyl}} = 5\,\frac{1}{\sigma_{compr}^{1/6}} \cdot \frac{\sigma_{0\ sph}}{\sigma_{0\ cyl}}$$

Calculated in accordance with this equation and plotted in Fig. 241 is the ratio $\frac{\sigma_{max\ sph}}{\sigma_{max\ cyl}}$ as a σ_{compr} function, for various values of a. From the graph it is clear that the $\sigma_{max\ sph}$ stresses exceed the

$\sigma_{max\ cyl}$ stresses 7-15 times for low σ_{compr} values and 1.1-2.5 times for high ones.

Let us compare the strength of cylindrical connections having linear contact (cases *1* and *2*, Fig. 242) with the strength of connec-

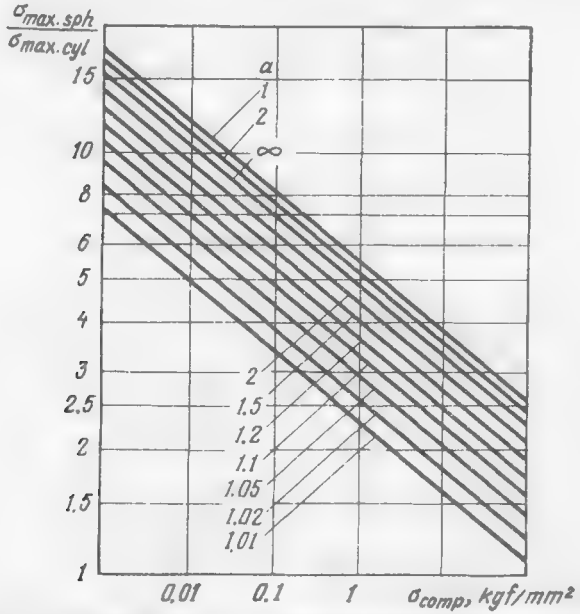


Fig. 241. Ratio of stresses $\sigma_{max\ sph}$ in spherical connections to stresses $\sigma_{max\ cyl}$ in cylindrical connections as a function of σ_{compr}

tions with surface contact (cases *3* and *4*). As in the case of spherical connections, assume that the compressive stress is equal to σ_{compr} =

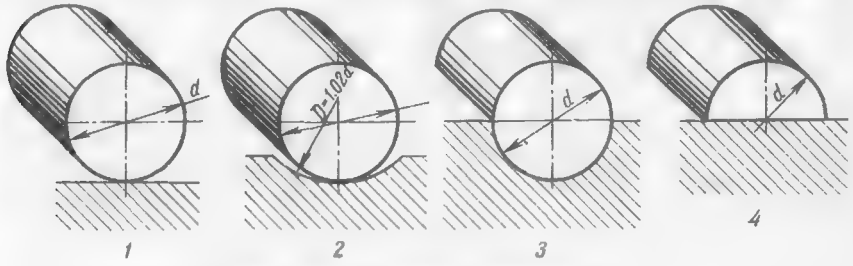


Fig. 242. Cylindrical connections

1 and *2* — point contact; *3* and *4* — surface contact

= 0.1 kgf/mm². For surface contact connections the bearing stresses are equal to the compressive stresses ($\sigma_{bear} = \sigma_{compr}$ = 0.1 kgf/mm²). The results of the comparison are given below

	Case 1	*Case 2*
σ_{max}, kgf/mm² .	30	4
$\sigma_{max}/\sigma_{bear}$	300	40
$\sigma_{max}/5\sigma_{bear}$ . . .	60	8

Clearly, the strength of connections, in which a cylinder rests on a flat surface is 60 times, and when a cylinder rests in a cylindrical socket with $a = 1.02$, is 8 times less than that with surface contact.

We now explain the influence of cylinder diameter on the strength of a connection.

Present Eq. (6.6) in the following way

$$\sigma_{max} = 87\sqrt{\sigma_{compr}}\,\sigma_0 = 87\sqrt{\frac{P}{bd^2}}\,\sigma_0 \qquad (6.7)$$

where $b = \frac{l}{d}$ —ratio of cylinder length to its diameter.

For geometrically similar cylinders b = const. Thus, it is evident from Eq. (6.7) that, other conditions being equal, the σ_{max} stresses are inversely proportional to the cylinder diameter.

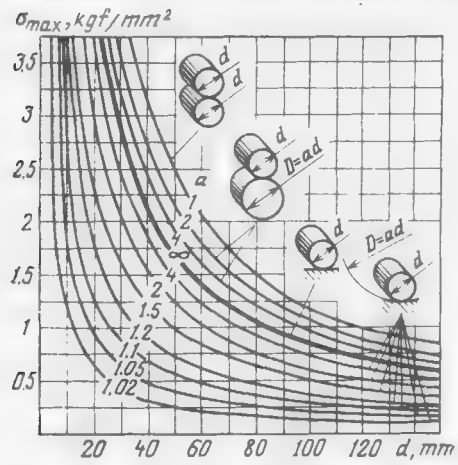


Fig. 243. Maximum stresses σ_{max} as a function of cylinder diameter d (with $P = 1$ kgf and $l/d = 1$)

The σ_{max} values given in Fig. 243, for cylinders of different diameters are calculated from Eq. (6.7). The general view of the σ_{max} curves is analogous to those of spherical connections (see Fig. 237), the only difference being that in the former case the stresses are smaller and the influence of diameter upon the strength is greater, than in the case of the spheres.

With assigned loads and cylinder diameters the load-carrying capacity of cylindrical connections can further be enhanced by making cylinders longer, i.e., by reducing the compressive stress σ_{compr}. This is impossible with spherical connections.

In practice, however, imperfections of manufacture cause irregular load distribution over the cylinder length as soon as the l/d ratio exceeds 1.5-2. As a result, loads are concentrated over cylinder rims.

Now, let us compare the strength of spherical (point contact) and cylindrical (linear contact) connections with that of surface contact connections (Fig. 244). Assume as unity the bearing stresses of a

Table 25

Stresses in Various Types of Connections

Type of connection	Stresses			
	σ_{max}, kgf/mm²	σ_{bear}, kgf/mm²	$\sigma_{max}/\sigma_{bear}$	$\sigma_{max}/5\sigma_{bear}$
Sphere in seat ($a=1.02$)	15	—	150	30
Cylinder in seat ($a=1.02$)	4	—	40	8
Surface contact ($a=1$)	—	0.1	1	1
Surface contact (cylinder with $l/d=1$)	—	0.0785	0.785	0.785

sphere seated in a socket with $a = 1$ ($\sigma_{bear} = 0.1$ kgf/mm²). The result of the comparison is given in Table 25.

Obviously, when $a = 1.02$, the strength of point-contact connections is 30 times, and linear-contact connections is 8 times less than

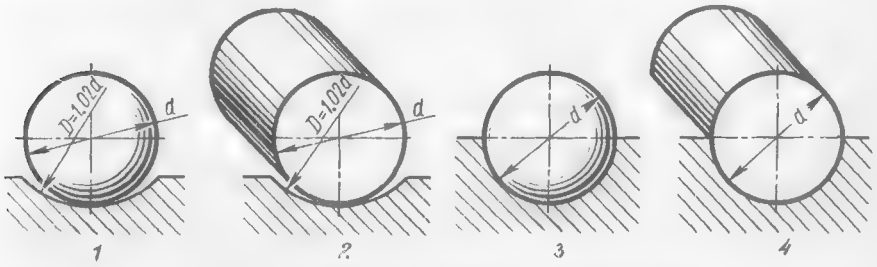


Fig. 244. Diagrams of contact in connections

1 — point contact; *2* — line contact; *3* — surface contact

the strength of connections having surfaces of a sphere and a plane in contact. In the case of a cylinder and plane surfaces in contact the respective figures are 40 and 10.

Hence, linear surface contact in terms of strength is more advantageous than point-contact (in the case considered it is approximately 4 times better), while surface contact is as many times better than the compared linear contact.

(*a*) *Design Rules*

The design rules for spherical and cylindrical connections carrying high loads are the following:

The contacting parts must be heat-treated to a hardness not lower than 60-62 Rc and processed to a surface finish not below ∇10 value.

With the aim of reducing contact stresses in the case when the working conditions for the connection allow the body accepting the

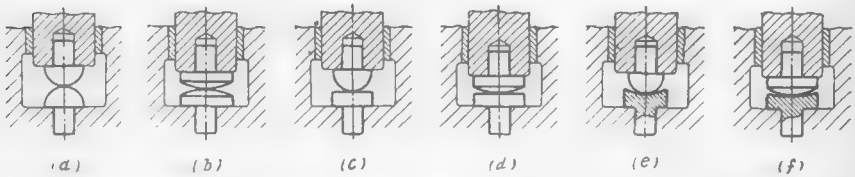


Fig. 245. Strengthening of a ball step bearing

load should be supported in a socket which has a diameter close to the body diameter ($D/d = 1.01\text{-}1.02$). An example of the progressive

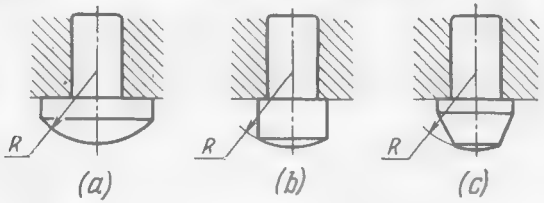


Fig. 246. Size reduction of a spherical working surface

improvement in strength of a spherical connection is illustrated in Fig. 245 (a spherical shaped contact unit). The most advantageous shape is a large diameter sphere seating in a spherical socket, Fig. 245*f*.

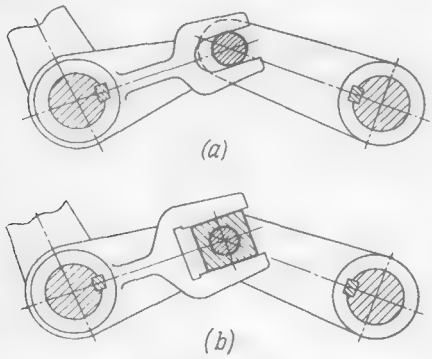


Fig. 247. Changing line contact to surface contact in a lever joint

In view of the fact that even in the case of the sphere and socket of large diameters having diameters close to one another the surfaces in contact are very small, thus, in the interest of reducing precision machining it is better to give the working surfaces the largest sizes which can be achieved by the manufacturing process (Fig. 246).

In all instances when this is possible in the design linear contact should be preferred to point contact and surface contact to linear.

As an example, let us consider a pin and two lever connection. The pin is fixed to one lever and slides in a fork of the other. The design,

shown in Fig. 247*a*, is irrational because contact on the rubbing surfaces is linear and the pin rapidly wears the fork. In the rational design (Fig. 247*b*) the pin is held in a block the sides of which slide in the fork. Thus, we have obtained surface contact between the pin and the block and also between the block and the fork. As a result the durability of the connection has been greatly increased.

Another example deals with a rotary pump blade (Fig. 248*a*). One end of the blade slides in a slot of rotor *1*, while the other end,

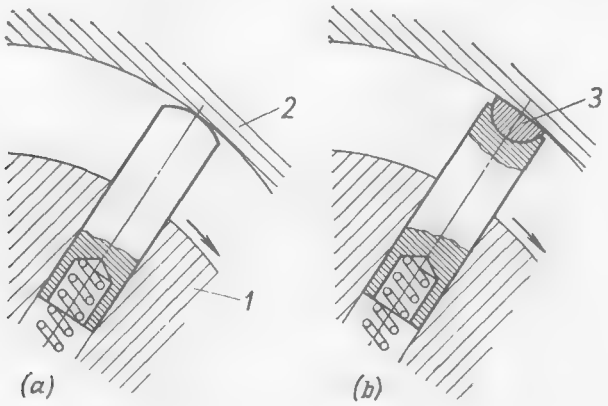


Fig. 248. Changing line contact to surface contact for a rotary pump vane

loaded with a strong spring and centrifugal force, is displaced relative to the eccentric stationary housing *2*. The contact between the blade and housing wall is linear; while moving the blade abrades the housing cylindrical surface. Operation conditions can be improved by introducing cylindrical insert *3* (Fig. 248*b*): thus, the contact between the insert and cylinder is a surface one and will materially decrease wear.

(*b*) *Connections Operating under Impact Loads*

The operating conditions in cyclically loaded connections are seriously impaired if the joint has clearance: coupling surfaces periodically move apart and close and shock loading occurs. When designed improperly, such a connection quickly becomes unserviceable due to overheating, work hardening and wear of surfaces.

To improve the work capacity of connections undergoing shock loads, it is advisable to:

raise the elasticity of the system by introducing shock-absorbers which mitigate the shock;

lessen stresses on working surfaces by changing point and linear contacts to surface contact and by increasing surface areas;

impart greater strength, hardness and thermal stability (e.g., by stelliting) to the working surfaces;

lessen or fully eliminate connection clearances;

provide ample lubricant to connections with the aim of creating shock-absorbing oil film, remove heat, emanating during impacts and (in the case of steel surfaces) eliminate tempering;

decrease weight of mechanism links to lower inertia loads.

For loads under control (e.g., in cam-actuated mechanisms) all measures must be taken to reduce the load and its degree of impact by decreasing the accelerations in the system (applying rationally-designed cams, e.g., parabolic and polynomial).

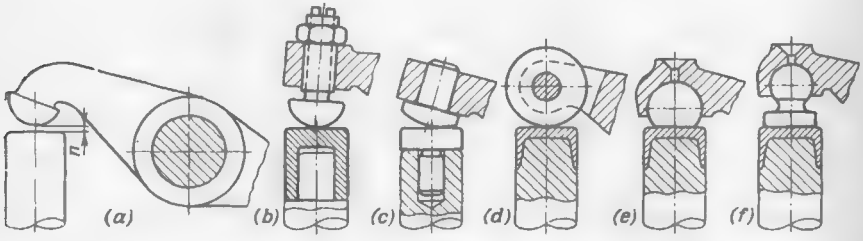


Fig. 249. Enhancing the operability of connections subjected to impact loads (a tappet actuated by a rocker)

An example of progressive strengthening of a reciprocating rod drive unit is displayed in Fig. 249. Since the clearance h is inescapable between contacting surfaces (Fig. 249*a*), the load has an impact character. The operating conditions are aggravated still more by the fact that owing to the kinematics of the system the movement of the rocker is accompanied by displacement of the striker over the rod end face.

The work capacity of the connection can be improved by the application of a screwed-in striker and rod end cap, hardened to a high degree (Fig. 249*b*). Disadvantage of the system is the point contact.

The connection operation can also be heightened by applying a striker with a large radiused cylindrical surface which ensures linear contact with lower contact stresses (Fig. 249*c*).

Still better is the design, shown in Fig. 249*d*, in which sliding friction is changed to rolling friction.

In the most rational design (Fig. 249*e*, *f*) the striker is made as a spherical insert with a flat working surface. Here the linear contact is changed to surface contact due to which pressures on the working surfaces are sharply lowered. Owing to its spherical shape, the connection acquires self-alignment which means uniform load distribution over the working surface regardless of possible distortions.

In the designs Fig. 249*d*, *f*, the rod end face is stellited. Stellite has high hardness (50-55 Rc) and keeps its hardness at elevated temperatures in contrast to hardened steels.

Fig. 250*a*, illustrates an example of how elasticity of a tappet can be enhanced. During the overloads the preset force of the spring is overcome mitigating the shock. The system is applicable only in the

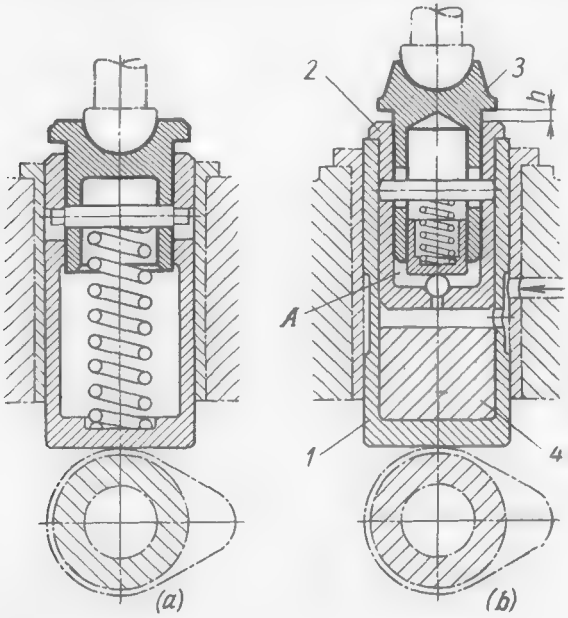


Fig. 250. Tappets
(*a*) spring-loaded; (*b*) hydraulic

cases with excess values of the driving force when some deviation of the motion may be permitted in the final link of the mechanism from the designed value given by the cam profile.

In addition to the increased hardness and decreased unit pressures on the working surfaces all measures should be taken to reduce connection clearances. Provision of adjustment (see Fig. 249*b*) allows minimum clearances, consistent with the mechanism correct functioning to be set and also compensation for wear. However, adjustments complicate use since periodic control of the mechanism state is required.

The best solution is automatic elimination of clearance. One of the ways is the application of hydraulic compensators.

Fig. 250*b*, illustrates a hydraulic tappet, used in internal combustion engines, in the valve drive mechanism.

The tappet consists of sleeve *1*, which reciprocates in a guide bush when actuated by a cam, linked by a rod to the valve drive mechanism. The tappet reverse stroke is actuated by a valve spring. In cylinder *2* there slides plunger *3* which has a spherical seat which receives the valve mechanism drive rod. A ball valve loaded by a light spring is arranged in cavity *A*. Oil from the pressure pipe of the engine is continuously fed into cavity *A* through a system of channels. Slightly raising the ball from its seat the oil enters the cavity and forces the plunger out of the cylinder, until the clearance *h* in all mechanism's links is eliminated. The plunger area is so calculated that the oil pressure cannot open the engine valve nor materially reduce the force, developed by the valve spring.

As the cam runs against the tappet, the pressure of oil in the space under the plunger increases, causing the ball valve to close. The driving force is transmitted through the column of oil, locked up in cavity *A*. Since oil is practically an incompressible medium, the mechanism acts as a solid system.

As the cam leaves the tappet the oil pressure under the plunger decreases and oil from the pressure pipe flows under the plunger compensating the loss which occurred during the tappet working stroke as oil leaked away through the clearances between the plunger and cylinder.

The system automatically assures mechanism operation without clearance regardless of thermal expansion and wear of the linking mechanisms working surfaces.

Clearly, oil leakage from under the plunger in no way affects the operation of the mechanism. Moreover, it is indispensible for correct functioning. If the system were sealed then a decreasing clearance in the joint, as a result of temperature drop (due to lower pressure, idle running, etc.), would probably cause incomplete valve closures. The plungers, moved out from the cylinders by an amount, corresponding to the preceding increased clearance, would be prevented from seating, thus keeping the valves a little open, which disturbs correct gas distribution. The oil leakage enables the mechanism to adapt itself to reduced clearances.

At the starting stages, when the pressure in the oil pressure pipe is absent, the system will operate for a short period with excessive clearances, because the driving force at this time is transmitted by the plunger thrusting against the tappet end face. Only after the oil pump has developed pressure, does the system come into action. To reduce the excessive clearance working period, it is reasonable to reduce the volume of the oil piping and also the volumes of oil spaces in the tappets, e.g., by using plungers made of light materials (plunger *4*, shown in Fig. 250*b*). To increase oil pump capacity is also feasible.

Chapter 7

Thermal stresses and strains

Elevated temperatures are observed not only in internal combustion engines, turbines, high-pressure compressors, etc., in which heating is a consequence of working processes. In "cold" machines the mechanisms operating at high speeds and under heavy loads (gears, bearings, cam-actuated mechanisms, etc.) are also heated. Parts undergoing cyclic loads under multiple "loading-unloading" cycles are also heated due to the elastic hysteresis.

Elevation of temperature is accompanied by changes in parts dimensions and may cause high stresses.

7.1. Thermal Stresses

Thermal stresses occur in the material if the latter cannot freely expand and contract under fluctuations of temperature. Such a behaviour is either the result of constraint (impediment) of thermal strains in a part by mated parts (contiguity constraint) or constraint of material fibres strain by contiguous fibres (configuration constraint).

(*a*) *Contiguity Constraint*

This kind of constraint can be illustrated by an example of rigid connection of several parts, which are either subjected to different temperatures in the course of operation or are made of materials possessing different coefficients of linear expansion.

Let stud *1* (Fig. 251) and bush *2* be made of materials whose coefficients of linear expansion are respectively α_1 and α_2, and $\alpha_2 > \alpha_1$. The temperatures of parts are respectively t_1 and t_2. When heated from some initial temperature (say, from zero) the stud and bush in the free state would lengthen by the amounts $l\alpha_1 t_1$ and $l\alpha_2 t_2$ (l—length of connection). In the restrained system the extension difference equals $l\,(\alpha_2 t_2 - \alpha_1 t_1)$ (in relative units $\alpha_2 t_2 - \alpha_1 t_1$), causes temperature interference. The thermal force P_t in the connection will cause (according to Hooke's law) lengthening of the stud, equal in relative units to P_t/E_1F_1, and contraction of the bush equal to

P_t/E_2F_2 (E_1 and E_2—are respectively Young's moduli of the materials; F_1 and F_2—cross-sections of the stud and bush).

Proceeding from the strain compatability law the sum of the above values must be equal to the amount of relative temperature interference

$$\frac{P_t}{E_1F_1}+\frac{P_t}{E_2F_2}=\alpha_2t_2-\alpha_1t_1$$

hence

$$P_t=(\alpha_2t_2-\alpha_1t_1)\frac{E_1F_1}{1+\frac{E_1F_1}{E_2F_2}} \quad (7.1)$$

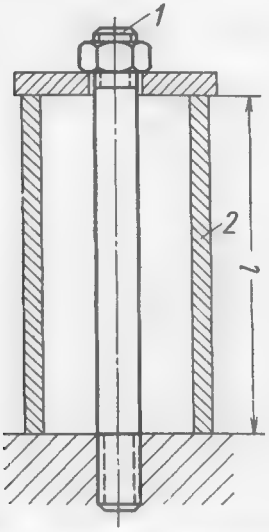


Fig. 251. Contiguity constraint diagram

According to Eq. (7.1) the tensile stress in the stud

$$\sigma_1=\frac{P_t}{F_1}=E_1(\alpha_2t_2-\alpha_1t_1)\frac{1}{1+\frac{E_1F_1}{E_2F_2}} \quad (7.2)$$

the compressive stress in the bush

$$\sigma_2=\frac{P_t}{F_2}=E_2(\alpha_2t_2-\alpha_1t_1)\frac{1}{1+\frac{E_2F_2}{E_1F_1}} \quad (7.3)$$

With the given value $\alpha_2t_2-\alpha_1t_1$ the stresses will be proportional to the factors

$$\sigma_{01}=\frac{E_1}{1+\frac{E_1F_1}{E_2F_2}} \quad (7.4)$$

$$\sigma_{02}=\frac{E_2}{1+\frac{E_2F_2}{E_1F_1}} \quad (7.5)$$

which can be termed as relative thermal stresses in the stud and bush, respectively.

The ratio

$$\frac{\sigma_{01}}{\sigma_{02}}=\frac{E_1}{E_2}\cdot\frac{1+\frac{E_2F_2}{E_1F_1}}{1+\frac{E_1F_1}{E_2F_2}}=\frac{F_2}{F_1} \quad (7.6)$$

i.e., is independent of the elasticity moduli of the stud and bush materials, but is completely dependent on the cross-section ratio of the parts.

Table 26

F_1/F_2 Ratio. Studs and Casings Have Equal Strength

Casing material	σ_{compr}, kgf/mm²	$F_1/F_2 = \sigma_{compr}/\sigma_b$
Steel 20Л	80	0.8
Grey cast iron СЧ32-52	100	1
High-strength cast iron ВЧ60-2	150	1.5
Aluminium alloy АЛ4	25	0.25

When $F_1/F_2 = 1$ the tensile stresses in the stud are equal to the compressive stresses in the bush ($\sigma_{01}/\sigma_{02} = 1$).

As a rule, the material of the stud is stronger than that of the bush (e.g., studs used to tighten together cast aluminium parts). Moreover, cast materials have generally higher resistance to compression than tension.

Let studs be made of quality steel with tensile strength $\sigma_b = 100$ kgf/mm² and casings of cast steel 20Л, grey cast iron СЧ32-52, high-strength cast iron ВЧ60-2 or aluminium alloy АЛ4 (Table 26).

If the studs and casings have equal strength (equal safety factor), their sections, according to Eq. (7.6), must be consistent with the ratio $F_1/F_2 = \sigma_{compr}/\sigma_b$.

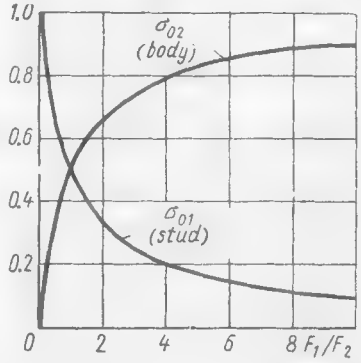


Fig. 252. Relative thermal stresses σ_{01} in the outer member and σ_{02} in the inner member as a function of their sections ratio F_1/F_2

In practice the cross sectional areas of castings are usually determined by their function and by the casting technology. Generally the strength of casings is many times that of fastening studs. Therefore when designing tightening connections it is necessary to begin from stud strength.

From Eq. (7.2) and Eq. (7.3) with given values of E_1 and E_2 thermal stresses in studs become higher with the decrease in the F_1/F_2 ratio (Fig. 252), approaching the value $E_1(\alpha_2 t_2 - \alpha_1 t_1)$ when $F_2 = \infty$ (absolutely rigid casing). In the casing thermal stresses grow with the

increase of the F_1/F_2 ratio approaching the value $E_2(\alpha_2 t_2 - \alpha_1 t_1)$ when $F_2 = 0$ (absolutely elastic casing).

On the graph, presented in Fig. 252, the values of σ_{01} and σ_{02} are given as a function of ratio F_1/F_2. Assumed as unity for σ_{01} is the

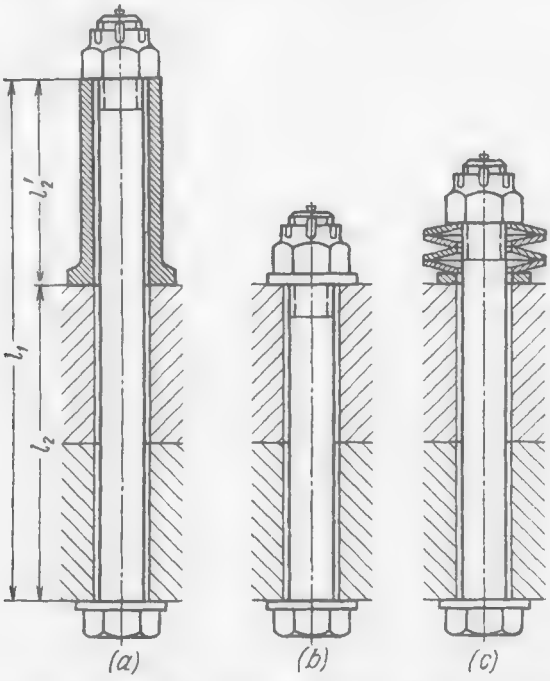


Fig. 253. Ways of reducing thermal stresses in tightened connections

value $E_1(\alpha_2 t_2 - \alpha_1 t_1)$, and for σ_{02} the value $E_2(\alpha_2 t_2 - \alpha_1 t_1)$. From the graph and Eq. (7.4), (7.5) it is possible to conclude that:

to reduce thermal stresses in studs it is necessary to make them *rigid* and the casing *elastic*;

to reduce thermal stresses in the casing studs have to be *elastic* and the casing *rigid*.

Since the strength of a casing is not decisive for the strength of the tightened connections therefore all thermally loaded connections must keep the following principle: *elastic casing—rigid studs*.

Equations (7.4-7.5) do not contain the length of the connection. This means that thermal forces and stresses are independent of the stud and bush lengths provided they are the same. Under similar conditions the theoretical values are identical, for example, in the case when a stud fastens a flange 10 mm thick or a casing 500 mm high.

In practice, the thermal stress value is affected by the elastic deformation of threads, washers, etc., which may decrease thermal stresses. This influence is relatively greater the shorter the studs. Elastic strains are connected with residual strains (distortion of threads and bearing surfaces). Short studs weaken in use quicker than longer studs which maintain their elasticity better.

Apart from thermal stresses, the strength of tightened connections is dependent on the preliminary tightening force as well as the working forces, acting upon the connection. The combination of these factors and practical recommendations on the design of fastening connections are considered further in Vol. 2, Section 1.

From the structure of Eq. (7.1), it is evident that thermal forces can be reduced by:

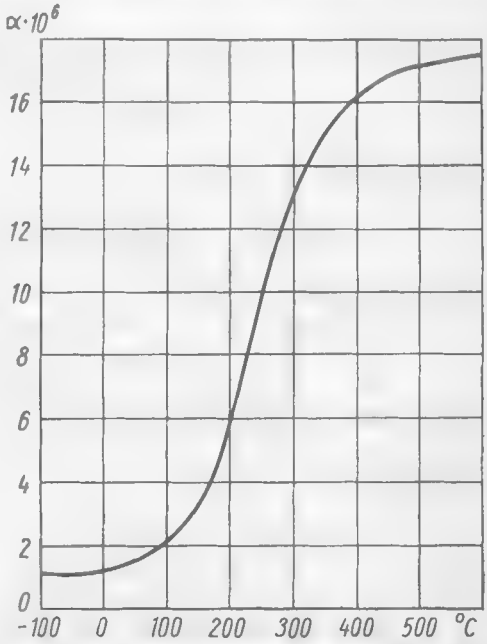


Fig. 254. Linear expansion coefficient of Invar as a function of temperature

decreasing the temperatures difference of parts to be joined together (e.g., cooling the part being fastened or increasing the temperature of the fastening part);

decreasing the differences in the linear expansion coefficients (by appropriate selection of materials for the parts to be mated).

If the materials of a fastened and fastening parts cannot be varied, then, thermal stresses can be lowered by introducing some additional elements (Fig. 253*a*) between the fastened and fastening parts made of a material with a lower linear expansion coefficient (e.g., invar).

Invar (H36) is an iron-nickel alloy (36% Ni). The coefficient of its linear expansion in the temperatures interval from minus 100°C to plus 100°C equals $\alpha = (1.1\text{-}2)\cdot 10^{-6}\ °C^{-1}$, and sharply rises with $t > 100°C$ (Fig. 254).

In this case the temperature interference

$$l_2\alpha_2 t_2 + l_3'\alpha_2' t_3' - l_1\alpha_1 t_1$$

where α_2, α_2' and α_1 = coefficient of linear expansion of respectively fastened, intermediate and fastening parts
t_2, t_2' and t_1 = their corresponding temperatures and lengths
and l_2, l_2' and l_1

Conditions for absence of temperature interference

$$l_2\alpha_2 t_2 + l_2'\alpha_2' t_2' - l_1\alpha_1 t_1 = 0$$

Introducing $l_2' = l_1 - l_2$ and assuming $t_2 = t_2' = t_1$, we find that

$$l_2' = l_2 \frac{\alpha_2 - \alpha_1}{\alpha_1 - \alpha_2'} \tag{7.7}$$

Substituting $\alpha_2 = 22 \cdot 10^{-6}$ (aluminium alloy), $\alpha_2' = 2 \cdot 10^{-6}$ (invar at $t < 150°C$) and $\alpha_1 = 11 \cdot 10^{-6}$ (steel), we find that

$$l_2' \approx 1.2 l_2 \tag{7.8}$$

i.e., to fully eliminate temperature interference, the length of an invar insert must be 20% greater than the length of parts being iastened. Such conditions are difficult to fulfil in practice.

There are some materials, whose coefficient of linear expansion is zero or even negative. In the latter case parts shrink when heated. Belonging to such materials are some sitalls ($\alpha = -2 \cdot 10^{-6}$). Calculations, based on Eq. (7.7) will give in this case $l_2' = 0.8 l_2$. Clearly, even under such conditions the intermediate bush must have substantial length.

In order to reduce thermal stresses, fastening bolts must occasionally be made of materials possessing high coefficients of linear expansion, for instance, of Cr-Ni austenitic steels (Fig. 253*b*), for which $\alpha = (14\text{-}18) \cdot 10^{-6}$. Let us compare fastening of an aluminium casing ($\alpha_2 = 22 \cdot 10^{-6}$) by means of bolts made of conventional structural steel ($\alpha_1 = 11 \cdot 10^{-6}$) and by bolts made of austenitic steel ($\alpha = 16 \cdot 10^{-6}$). When changing to the austenitic steel the thermal stresses decrease in the ratio

$$\frac{\alpha_2 - \alpha_1}{\alpha_2 - \alpha_1'} \cdot \frac{22 - 11}{22 - 16} = 1.84 \tag{7.9}$$

i.e., approximately by half.

It should be remembered that the strength of austenitic steels is significantly lower than high-grade structural steels. For example, austenitic steel, grade ЭИ69, with $\alpha = (16\text{-}18) \cdot 10^{-6}$ has the tensile strength level $\sigma_b = 70$ kgf/mm^2 and yield limit $\sigma_{0.2} = 40$ kgf/mm^2. Structural steel, grade 30ХГС ($\alpha = 11 \cdot 10^{-6}$) extensively used for producing load-carrying bolts has: $\sigma_b = 110$ kgf/mm^2 and $\sigma_{0.2} =$

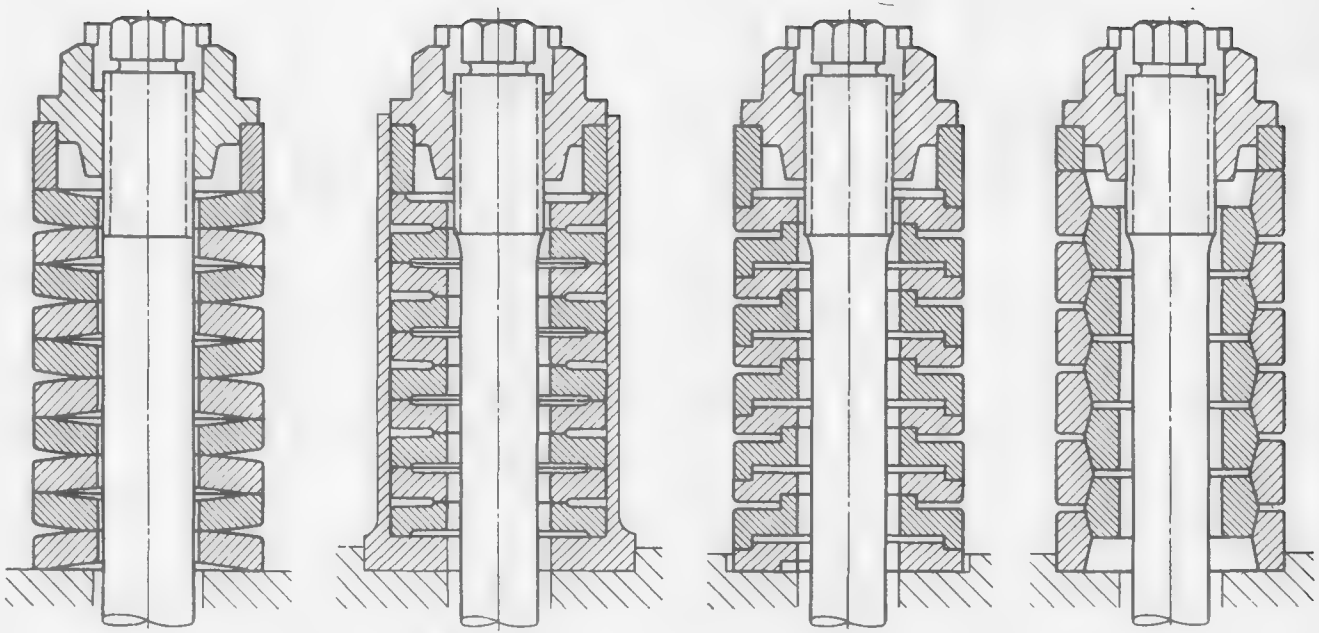


Fig. 255. Design of elastic elements in tightening systems

$= 85$ kgf/mm^2. From the strength point of view the latter steel, for the given case, is definitely more advantageous in spite of its small coefficient of linear expansion.

In addition, austenitic steels are much more expensive than low- and medium-alloy steels.

An efficient means of reducing thermal stresses in bolts is to enhance elasticity of the part being fastened (see Eqs. 7.2 and 7.3). If, however, this is impossible, then some spring elements (Fig. 253*c*) must be placed under the bolts, which is equivalent, in effect, to increasing the casing elasticity. If the spring element has sufficient elasticity, the thermal forces can practically be fully cancelled, the spring element considered as a preliminary tightening force and as a working force, acting on the connection.

This method is often applied to absorb thermal deformations when several details are assembled on one shaft if the details are made from alloys having high linear coefficient of expansion (e.g., rotors of multi-stage axial compressors). To fix and mount such details requires considerable axial forces. That is why in a given case spring elements are made as a stack of numerous, strong and very rigid elements (Fig. 255), which when summed give the required elasticity. Elastic element calculations are described in Vol. 2, Section 1.

(*b*) *Form Constraint*

Thermal stresses, produced by form constraint, occur during non-uniform heating of the part, when individual fibres of material cannot (due to the shape of the part) expand in accordance with the thermal expansion law. In contrast to the contiguity constraint, stresses here are due only to the existence of a temperature gradient in the part (with stationary heat flow, when heat flows from hot areas to cooler ones, or during unstable heat flow, e.g., thermal shock, when a wave of heat propagates throughout the part).

As a rule, hot areas of a part, having temperatures exceeding the average, undergo compressive stresses and the more cool areas—tensile stresses.

Thermal stresses do not occur in a uniformly heated part, where the temperature at any portion is the same.

(*c*) *Flat Walls*

Imagine a flat wall of thickness s (Fig. 256*a*), through which passes, normally to its plane, a uniform thermal flow. Assume also that the wall surface facing the heat source has temperature t_1, and that the opposite surface temperature is t_2 and that $t_1 > t_2$. The temperature across the wall according to the heat transfer theory in this case

changes to a linear law. The average wall temperature

$$t_{av} = \frac{t_1 + t_2}{2} \tag{7.10}$$

Now imagine that the plate consists of a number of thin parallel layers. If each layer could expand freely when heated, then the layers with temperature above t_{av} would expand relative to the average

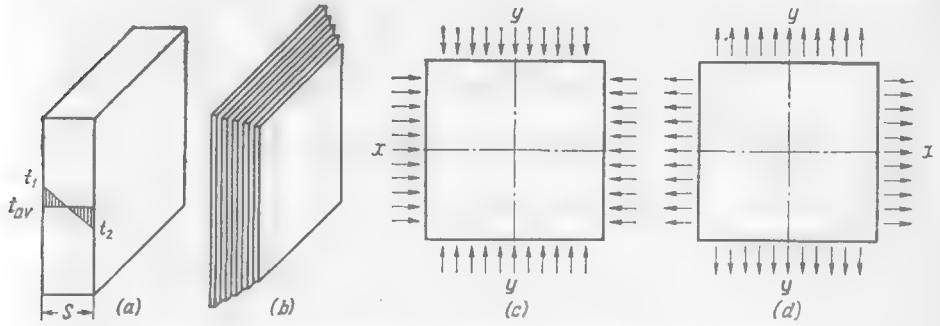


Fig. 256. Determination of thermal stresses in a flat wall

layer, and the layers with temperatures below t_{av} would shrink. Hence, the plate would take the shape, pictured in Fig. 256*b*.

The relative extension of the extreme, most heated layer

$$e_1 = \alpha (t_1 - t_{av}) = \alpha \left(t_1 - \frac{t_1 + t_2}{2} \right) = \alpha \frac{t_1 - t_2}{2} \tag{7.11}$$

The relative shrinkage (shortening) of the coolest layer

$$e_2 = \alpha (t_{av} - t_2) = \alpha \left(\frac{t_1 + t_2}{2} - t_2 \right) = \alpha \frac{t_1 - t_2}{2} \tag{7.12}$$

i.e.,

$$e_2 = e_1 = e_{max} = \alpha \frac{t_1 - t_2}{2} \tag{7.13}$$

If the plate maintains its flat shape during heating, then all its layers by virtue of deformation compatability, must have sizes equal to the sizes of the average layer. In such a plate the most heated layers are compressed by the constraint of adjacent less heated layers (Fig. 256*c*), while the coolest layers are stretched by the action of the hot layers (Fig. 256*d*), each along two, mutually perpendicular directions. The highest stresses arise in the extreme surface layers.

As known from the theory of elasticity, relative elongation under the conditions of a biaxial stressed state

along axis x

$$e_x = \frac{\sigma_x}{E} - m \frac{\sigma_y}{E} \tag{7.14}$$

along axis y

$$e_y = \frac{\sigma_y}{E} - m\frac{\sigma_x}{E} \tag{7.15}$$

where σ_x = stresses along x and y axes respectively
and σ_y
E = Young's Modulus
m = Poisson's ratio (transverse strain coefficient), which is, in fact, the ratio between the value of transverse compression and longitudinal elongation within the limits of elastic strains, taken for the case of simple tension in one direction

With symmetrical tension-compression (as in the case under consideration)

$$\sigma_x = \sigma_y = \sigma; \quad e_x = e_y = e$$

Consequently

$$\sigma = Ee\frac{1}{1-m}$$

Substituting the e-value from Eq. (7.13) into the above expression, gives the maximum stress value in the extreme layers

$$\sigma_{max} = \pm E\alpha \frac{1}{1-m} \cdot \frac{t_1 - t_2}{2} \tag{7.16}$$

where the plus sign refers to tension, and the minus sign to compression.

The stresses across the wall are distributed according to a linear law similar to the temperature change law.

The temperature gradients may be expressed in terms of the quantity of heat Q, passing through a wall in unit time and unit surface. According to Fourier

$$Q = \frac{\lambda}{s}(t_1 - t_2) \text{ cal/m}^2\cdot\text{h} \tag{7.17}$$

where λ = heat conductivity coefficient of the material, cal/m·h·°C
s = wall thickness, m

Substituting $t_1 - t_2$ from Eq. (7.17) into Eq. (7.16), gives

$$\sigma_{max} = \frac{Q}{2} s \frac{E\alpha}{\lambda} \cdot \frac{1}{1-m} \tag{7.18}$$

(*d*) *Thermal Strength of Materials*

As obvious from Eq. (7.18) the maximum thermal tension-compression stresses for a given heat flow intensity are proportional to the wall thickness and factor $\frac{E\alpha}{\lambda} \cdot \frac{1}{1-m}$, which is specific for any material (Table 27).

Table 27

Thermal Strength Characteristics of Materials

Materials	$E \cdot 10^{-3}$ in kgf/mm²	$\alpha \cdot 10^{6}$ in °C⁻¹	λ in cal/m·h·°C	m	$\frac{E\alpha}{\lambda} \cdot \frac{1}{1-m}$	σ_b in kgf/mm²	$\frac{\sigma_b \lambda (1-m)}{E\alpha}$
Grey cast irons	8	11	35	0.15	3	30	10
Garbon steels	21	11	40	0.3	8.3	60	7.2
Alloy steels	21	12	35	0.3	10.4	120	11.5
Stainless austenitic steels	21	16	15	0.3	32	70	2.2
Aluminium alloys — castable	7.2	22	150	0.33	1.6	20	12.5
Aluminium alloys — wrought						50	31
Magnesium alloys — castable	4.2	28	70	0.33	2.5	15	6
Magnesium alloys — wrought						25	10
Bronzes	11	18	70	0.33	4.2	60	14
Titanium alloys	12	8.5	7	0.3	21	120	5.7

Materials, with a zero linear expansion coefficient are free from thermal stresses due to form constraint.

Fig. 257 gives the values of linear expansion coefficient for various materials, plotted against temperatures.

For all materials the $\frac{1}{1-m}$ factor is close to 1.5 (except cast irons, for which the factor equals 1.18). When roughly comparing the materials in terms of thermal stresses, a simplified expression $\frac{E\alpha}{\lambda}$ may be employed.

A comparative study of Table 27 clearly shows that according to the thermal stress values $\left(\text{low valued factor } \frac{\alpha}{\lambda} \cdot \frac{1}{1-m}\right)$ light alloys are the most advantageous and titanium alloys and stainless steels the worst.

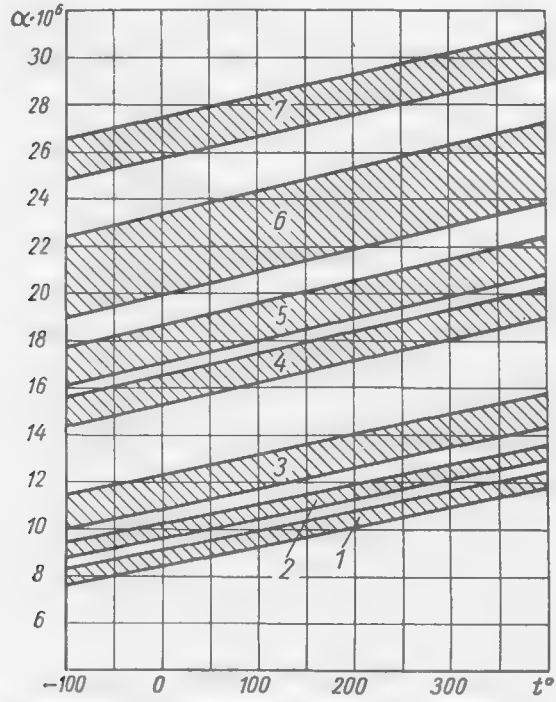


Fig. 257. Linear expansion coefficient of metals as a function of temperature

1 — titanium alloys; *2* — martensitic steels; *3* — pearlitic steels and cast irons; *4* — austenitic steels; *5* — copper-base alloys; *6* — aluminium alloys; *7* — magnesium alloys

Thermal strength, i.e., material resistance to the action of thermal stresses, can be expressed by the ratio between the tensile strength σ_b of the material and the factor $\frac{E\alpha}{\lambda} \cdot \frac{1}{1-m}$ (analogous to safety strength)

$$n = \frac{\sigma_b \lambda (1-m)}{E\alpha}$$

The values of this factor are presented in the extreme right-hand column of Table 27.

By their resistance to thermal stresses (high value of the factor) wrought aluminium alloys occupy the first place. Much less profitable are titanium alloys and austenitic stainless steels.

The above cited correlations are valid only for temperatures up to 200°C when the characteristics of strength, elasticity, linear expansion and heat conductivity for usual structural materials undergo comparatively small changes. These correlations become invalid at higher temperatures, since now it is the heat-resistance, that is of the most important significance Belonging to heat-resistant materials are steels, alloyed Ni, W, Mo, Ta, nickel-based alloys, titanium alloys, etc. In the high-temperature domain the qualitative correlations between materials alter. With the increase of temperature most of the above-examined materials (e.g., steels of conventional composition) lose their strength; some of them are unable to endure high temperatures at all (light alloys). Conversely, titanium alloys, which under moderate temperatures are hardly not the worst, according to their thermal stress values, now, due to their high heat-resistance take one of the first places.

(*e*) *Curvilinear Walls*

In our previous argument it was assumed that a flat plate subjected to thermal strains preserved its flatness, that it was either arranged in rigid guide-ways or had sufficient rigidity against bending. Now, if the plate is freely deformed under the influence of the temperature gradient, then the thermal strains will decrease, completely vanishing under certain conditions, for example, if the plate is thin enough, made of a material with low elasticity and can bend completely so that its external fibres elongate and internal ones shorten by the value $\alpha (t_1 - t_2)$. In this case the plate will bend to a sphere (Fig. 258*a*), whose mean radius

$$R = s\left[\frac{1}{\alpha (t_1 - t_2)} + 1\right]$$

Should free bedning be possible in only one direction, then the plate will bend to a cylinder (Fig. 258*b*), whose mean radius

$$R_{mean} = \frac{s}{\alpha (t_1 - t_2)}$$

Stresses along y-axis in this case will weaken or vanish ccmpletely, while stresses along the x-axis will maintain. The amount of these stresses can be found by putting $\sigma_y = 0$ in Eq. (7.14).

Then

$$e_x = \frac{\sigma_x}{E}$$

Since according to Eq. (7.13)

$$e_x = \alpha \frac{t_1 - t_2}{2}$$

then

$$\sigma_x = E\alpha \frac{t_1 - t_2}{2} = \frac{Q}{2} s \frac{E\alpha}{\lambda} \tag{7.19}$$

Hence, Eq. (7.18) gives the maximum value of thermal stresses in a wall, which is unable to change its shape, while Eq. (7.19) the

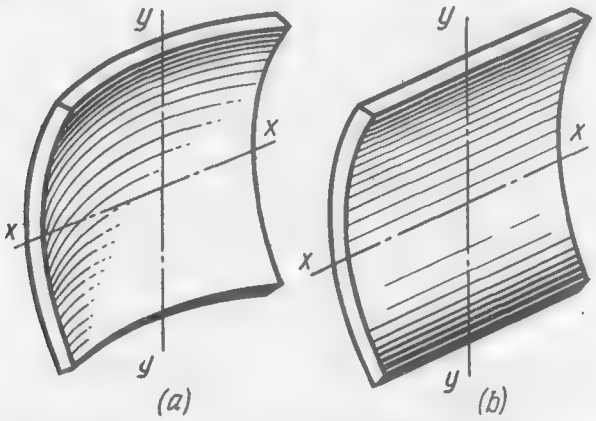


Fig. 258. Flexure of a thin plate under the effect of thermal stresses

value of stresses, arising when the wall shape can change in one direction only. In the intermediate cases the thermal stress values will vary from 1 to $\frac{1}{1-m}$ (i.e., from 1 to 1.5 on average).

(f) *Cylindrical Tubes*

In practice there are cases when the part due to its configuration maintains its shape fully or nearly in the presence of a temperature

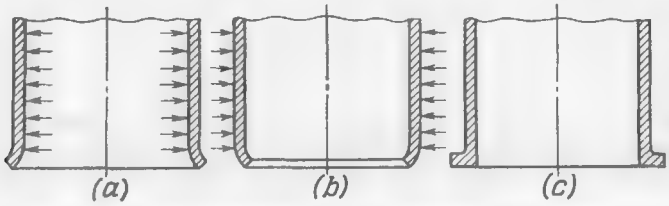


Fig. 259. Deformation of an unrestrained cylinder end

gradient. A typical example is a cylindrical tube of a sufficiently large length. In the event of a unilateral heating, for example, from inside, Fig. 259*a*, the tube will expand both radially and axially, keeping, on the whole, its cylindrical shape. The internal, warmer wall layers,

will be subjected to compressive stresses, and those on the outside, the cooler layers, to tensile stresses.

The stresses will fall only at the free end of the tube, where the constraining influence of the circular sections is weaker as a result of which the tube expands in a flare-like way.

The picture is quite the opposite when the tube is heated from outside (Fig. 259*b*): the outside, warmer layers are subjected to compression, and the inside cooler ones, to tension. The tube free ends "flare" inwards.

When it is necessary to preserve the true cylindrical shape circular rigidity ribs should be introduced on the tube ends (Fig. 259*c*).

The correlations, formulated for a flat wall, remain valid for a cylindrical tube as well, provided due corrections are made for curvilinearity of walls and different distribution of temperature across the wall.

If temperature distribution across the wall is arbitrary (Fig. 260), the average wall temperature will be

$$t_{av} = \frac{1}{s} \int_0^s t\, dz$$

being, in fact, the height of a rectangle with base s, area $abcd$, having a value equal to the area of the temperature diagram.

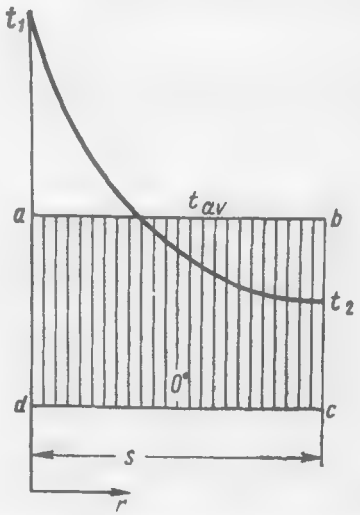


Fig. 260. Determination of the average temperature of a wall, with the temperature across the wall changing nonuniformly

When a stationary heat flux is flowing outwardly, the temperature across a cylindrical wall changes according to a logarithmic law

$$t = t_1 = \frac{Qr \ln R}{\lambda}$$

where Q = heat quantity, flowing through the wall in unit time
λ = heat conductivity coefficient
r and R = respectively internal and external radii of the cylinder

For the case under consideration the average temperature of wall

$$t_{av} = \frac{2}{s(R+r)} \int_r^R t\rho\, d\rho$$

where ρ is the present radius.

The t_{av} value is determined either analytically or graphically.

The maximum thermal stresses in the extreme layers of the cylindrical wall are similar to Eq. (7.16)

$$\sigma_{max} = \pm E\alpha \frac{1}{1-m} \cdot \frac{t_1 - t_2}{2} c$$

where c is the correction factor for the wall cylindricity.

For tensile stresses (wall cold side)

$$c = \frac{2\gamma^2}{\gamma^2 - 1} - \frac{1}{\ln \gamma}$$

for compressive stresses (wall hot side)

$$c = \frac{2}{\gamma^2 - 1} - \frac{1}{\ln \gamma}$$

where $\gamma = \frac{R}{r}$.

For small wall thicknesses the effect of curvilinearity can be neglected, and the thermal stresses determined from Eq. (7.16).

Thermal stresses may reach substantial values, and in certain cases can limit the strength of a part.

We now consider a steel tube heated from outside ($I.D.$ = 100 mm, $O.D.$ = 120 mm) and having a wall-to-wall temperature difference of 30°.

According to Eq. (7.16) thermal stresses for a flat wall

$$\sigma_{max} = E\alpha \frac{1}{1-m} \cdot \frac{t_1 - t_2}{2} = 22 \cdot 10^3 \cdot 11 \cdot 10^{-6} \cdot 1.44 \cdot 15 = 5.5 \text{ kgf/mm}^2$$

The correction factor c:

for tension

$$c = \frac{2 \cdot 1.2^2}{1.2^2 - 1} - \frac{1}{\ln 1.2} = \frac{2.88}{0.44} - \frac{1}{0.182} = 6.55 - 5.5 = 1.05$$

for compression

$$c = \frac{2}{1.2^2 - 1} - \frac{1}{\ln 1.2} = \frac{2}{0.44} - \frac{1}{0.182} = 4.55 - 5.5 = -0.95$$

Hence, the tensile stresses

$$\sigma_{tens} = 5.5 \cdot 1.05 = 5.8 \text{ kgf/mm}^2$$

compressive stresses

$$\sigma_{compr} = 5.5 \cdot 0.95 = 5.2 \text{ kgf/mm}^2$$

The thermal tensile stresses obtained are as if the tube were subjected to a burst of internal pressure, equal to (when calculating the tube wall in tension to the formula of Boyle-Mariotte)

$$p = 10^2 \cdot 5.8 \frac{2s}{d} = 10^2 \cdot \frac{5.8 \cdot 20}{100} = 116 \text{ atm}$$

(*g*) *Addition of Thermal and Working Stresses*

Generally thermal stresses are associated with stresses, produced by outer loads. Such a combination can be beneficial, if the thermal and working stress association results in lower stresses and can also be unfavourable when the resultant stresses are higher. This depends on the relationship between the thermal and working stresses and the law of their change through the wall.

Figure 261*a*, shows a thin-walled tube, carrying a hot fluid or a gas under high pressure and cooled from outside (continuous arrows show the direction of pressure, while dashed arrows indicate the direction of heat flow). Distribution of working stresses across the wall is shown according to the Boyle-Mariotte formula, with a straight line. The summation of working σ_w and thermal σ_{th} stresses produces the tensile stress peak σ on the outside (Fig. 261*a*, III).

For a tube, carrying an operating fluid or a gas under high pressure and heated from outside (Fig. 216*b*), the summation of working σ_w and thermal σ_{th} stresses produces the tensile stress peak σ on the wall inside.

If the tube is subjected to an external pressure, then heating both from inside (Fig. 261*c*) and outside (Fig. 261*d*) causes only the compressive stress peaks, which are less harmful than tensile stresses.

In thick-walled tubes the distribution of working and thermal stresses across the wall is different. Here with favourable relationship thermal stresses may decrease the resultant stresses and lead to a more uniform distribution of stresses across the wall (Fig. 261*e*).

It should be emphasized, however, that an increase in wall thickness does not always imply an improvement of tube strength in terms of its resistance against the joint action of external and thermal loads.

According to the Boyle-Mariotte formula, the tensile stresses, produced by internal pressure, for thin-walled tubes will equal

$$\sigma_w = \frac{pd}{2s}$$

decreasing with the increase of wall thickness. The thermal stresses, as seen from Eq. (7.19), at a given heat flow intensity Q, will rise with the increase of wall thickness. Consequently, there exists some optimal wall thickness at which the summed stresses across the wall have a minimum value.

Determining thermal stresses σ_{th} from Eq. (7.18), one obtains the following relation for total stress

$$\sigma = \sigma_w + \sigma_{th} = \frac{pd}{2s} + \frac{Qs}{2} \cdot \frac{E\alpha}{\lambda} \cdot \frac{1}{1-m}$$

Differentiating this relation with respect to s and equating the derivative to zero, the maximum value of s, for which σ has its

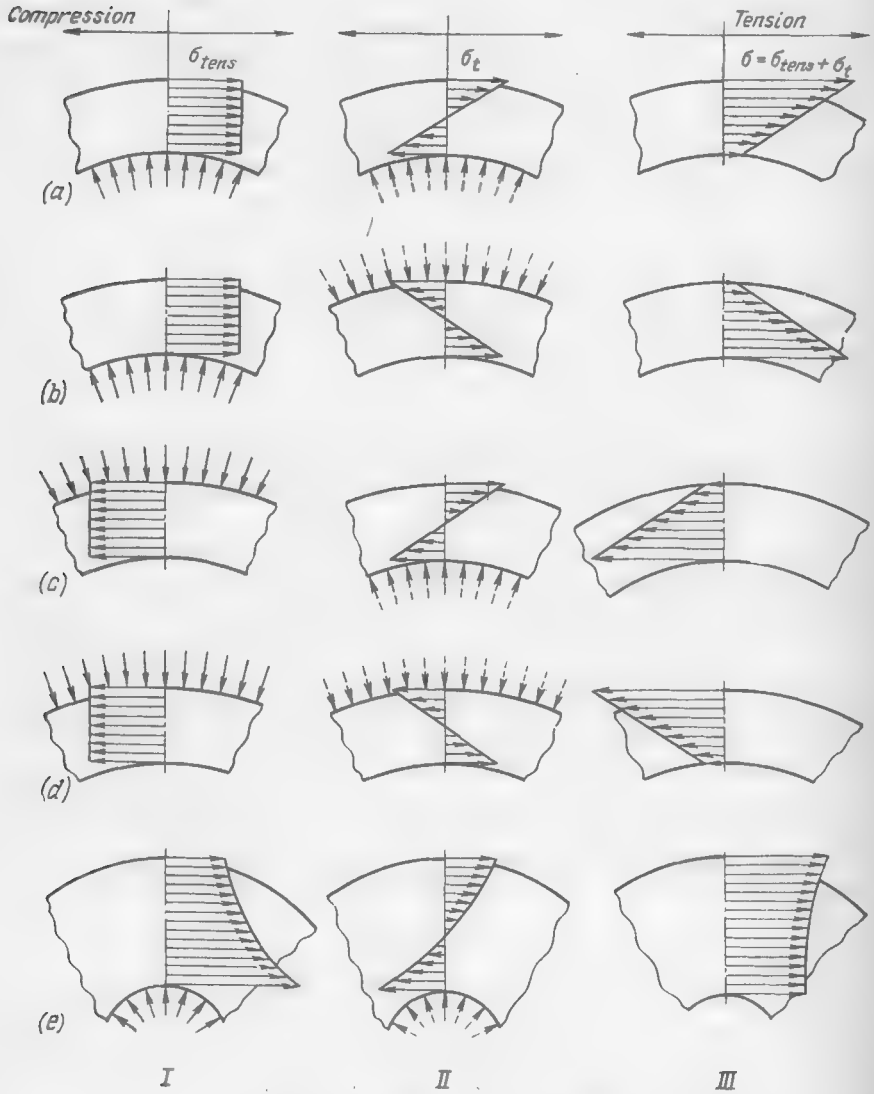


Fig. 261. Stresses in tube walls

I — pressure-induced; *II* — thermal; *III* — summed

minimum value will be found:

$$s = \sqrt{\frac{pd}{Q}} \sqrt{\frac{\lambda}{E\alpha} \frac{1}{1-m}} \tag{7.20}$$

It can be seen that this value depends on the operational fa tors (Q, p) and also upon the characteristics of the material (λ, E, α and m).

Presented in Fig. 262 as a function of wall thickness are the stresses in a steel tube, having the following parameters: $d = 100$ mm, $p = 100$ kgf/cm² and $Q = 100\,000$ cal/m²·h. The total stresses will have a sharply defined minimum at a wall thickness of $s \approx 10$ mm. Increasing wall thickness beyond this optimal value leads to a stress increase.

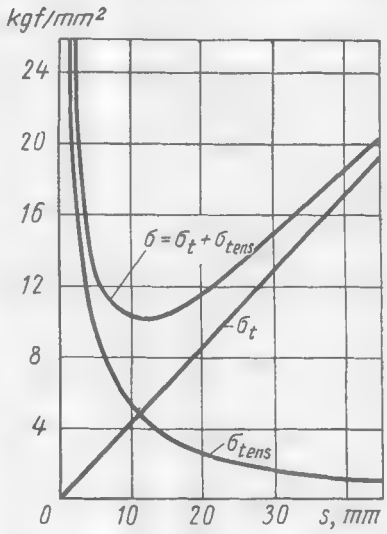


Fig. 262. Optimum wall thickness of a tube subjected to tensile stresses σ_{op} and thermal stresses σ_t

(*h*) *Disk-Like Parts. Rotors*

Thermal stresses play a very important role in the strength of high-speed rotors of thermal machines (e.g., turbines, centrifugal and axial compressors, etc.). The rotors, being subjected to bursting loads produce centrifugal forces and thermal stresses caused by uneven temperatures in the rotor body. Generally the temperature is higher at the rotor periphery. Here thermal compressive stresses occur. At the hub, i.e., where the tensile stresses from the centrifugal force have their greatest value, arise thermal tensile stresses. Additional tensile stresses are added for interference fitted rotors.

The distribution of thermal, centrifugal and total stresses in the symmetrical cross-sectional plane of the rotor is shown diagrammatically in Fig. 263*a*.

Adding the thermal stresses σ_{th} and centrifugal stresses σ_{ctf} shows a tensile stress peak at the rotor hub.

To determine the thermal stresses in the rotor is difficult since the regulating of the changing temperature in the rotor body depends on the operating conditions. Moreover, rotors in the majority of cases have intricate profiles which affect the magnitude of thermal stresses in the axial and circumferential directions.

Rather critical is the starting period when the blades and periphery of rotor quickly heat under the influence of operating gases,

while the hub still remains cold. In this case the tensile stresses near the hub are maximum. Under working conditions the rotor temperature equalizes due to which thermal stresses decrease.

When the blade temperatures fall, during idle running, the picture is the opposite: the rotor periphery becomes cooler than the hub

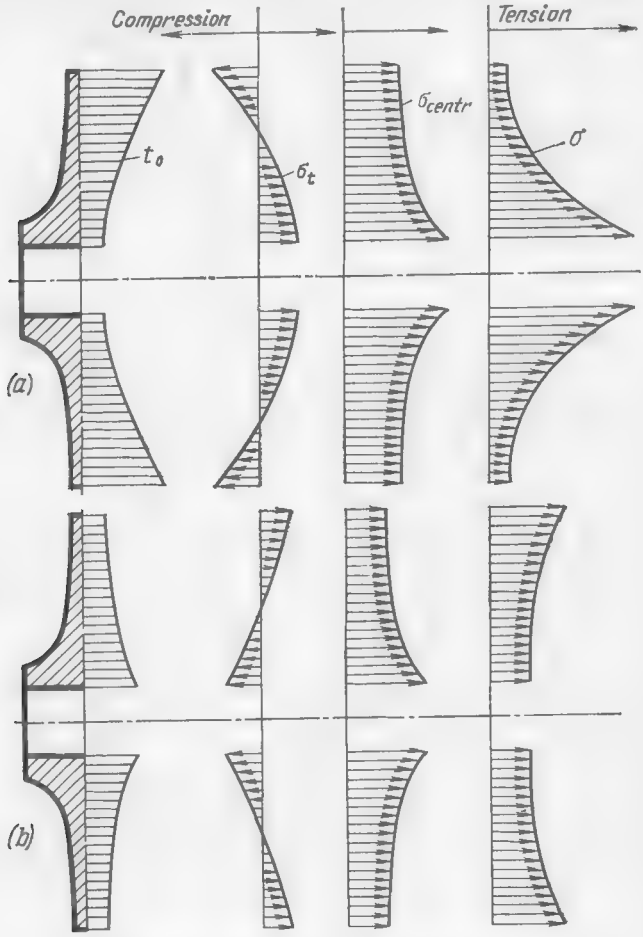


Fig. 263. Stresses in a rotating disk

t° — temperature; σ_t — thermal stresses; σ_{centr} — stresses due to centrifugal forces; σ — summed stresses

(Fig. 263*b*), and at the periphery occur thermal tensile stresses and at the hub, compressive ones. The peak of the total stresses then shifts to the periphery. Since rotor speed is less during idle running these working conditions are less harmful to rotor strength.

(*i*) *Decreasing Thermal Stresses*

Methods for decreasing thermal stresses being caused by configuration constraint, are aimed primarily at eliminating or, at least, diminishing the source of trouble-irregularities of temperature field over the part section. Sometimes this is obtained by cooling the part. Thus, for turbine rotors it is advisable to cool their peripheral parts. Cooling of the central portion of rotor is irrational as, when working, the drop of temperature can cause even greater bursting stresses in the hub.

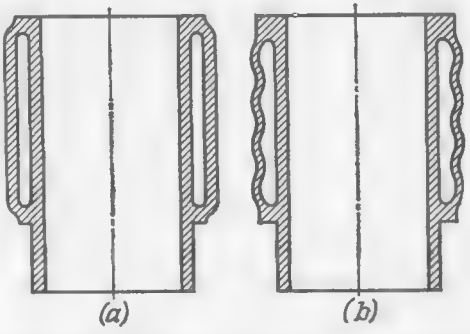


Fig. 264. Reduction of thermal stresses through introduction of a thermal buffer

If the temperature differential cannot be eliminated owing to some functional reason (e.g., heat-exchanger tubes), then it is advantageous to use materials which have beneficial combinations of strength, heat-conductivity and thermal expansion characteristics (see Table 27). For instance, sitall tubes with a zero coefficient of linear expansion are perfectly immune to thermal stresses.

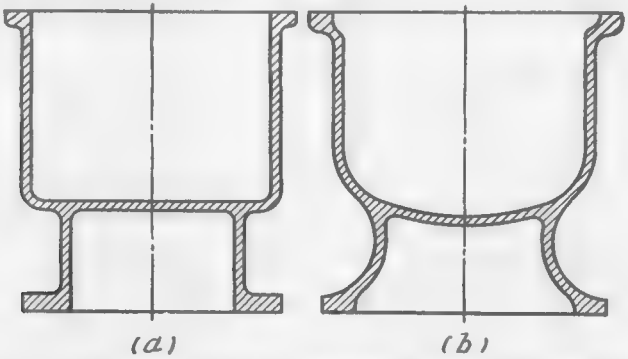


Fig. 265. Reduction of thermal stresses by smoothening the shape

Thermal stresses can be broken down by using thermal buffers, i.e., by imparting compliance to areas where the temperature differs from the temperature in adjacent regions.

Figure 264 shows ways of improving the design of an engine cylinder cooling jacket. In the rigid design (Fig. 264*a*) significant thermal stresses are possible due to temperature differences between the jacket and cylinder walls. In the cylinder walls which have the

higher temperature axially directed compressive stresses arise and in the jacket walls tensile stresses. The corrugated jacket walls, materially increasing the elasticity of the system, Fig. 264*a*, sharply reduce the thermal stresses, Fig. 264*b*.

In box-shaped components stiffening collars, flat partitions and abrupt transitions (Fig. 265*a*), which enhance the configuration constraint factor, should be avoided. More beneficial are conical, spherical and other similar shapes, which assure smooth transitions from one part of the detail to another part (Fig. 265*b*). These measures help to level the temperature gradient and increase the compliance of parts in the direction of acting thermal forces.

(j) *Expansion Joints*

In several cases without detriment to the functional purpose of the part is complete elimination, or almost complete elimination of

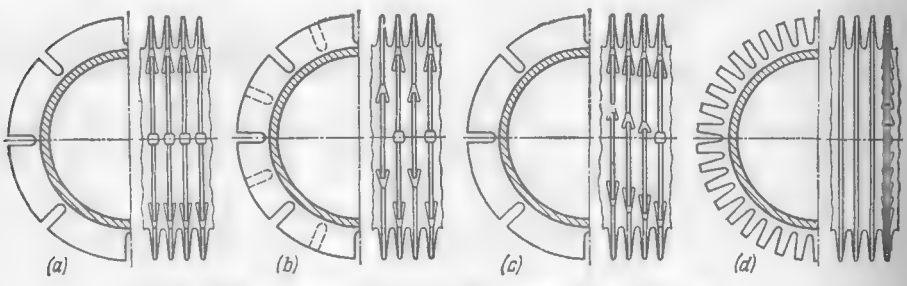


Fig. 266. Expansion joints in circular radiator fins

form restraint as the original source of thermal stresses. Expansion joints, which serve as an example are, in fact, radial grooves made in the circular fins of sleeves in air-cooled engines (Fig. 266*a*). To avoid any distortion of correct cylindrical shape such grooves are arranged either in a staggered (Fig. 266*b*) or helical (Fig. 266*c*) pattern.

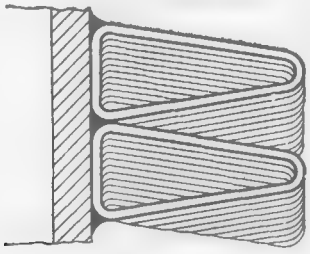


Fig. 267. Cooling surface formed by coiled spirals

Expansion joints do not significantly worsen heat-dissipation of ribs.

The ribbing can be totally deprived of thermal stresses if the grooves are arranged so thickly that the circular ribs turn into separate pillars (needle-like surface), like those pictured in Fig. 266*d*. Loss of the cooling surfaces at the

areas of grooves is compensated by the new cooling surfaces formed at the groove faces. Compensation can be complete if the groove width equals rib thickness. Moreover, thermal dissipation is still better owing to the enhanced turbulenc of air flow amidst the ribs. The design is considerably lighter, approximately by half, when the groove width equals that of the cooling needles.

Still further development of this cooling principle is in the creation of a brush-like surface, for example, by soldering spiral wire coils to the walls (Fig. 267).

7.2. Thermal Strains

Thermal strains sometimes change completely the size of parts and their relative position in a unit. This feature must be accounted for when designing units, comprising parts having different working temperatures or made of materials possessing different linear expansion coefficients.

(a) *Axial Clearances*

Thermal strains may to a large degree change axial clearances in mechanical joints.

Let us take for an example the design of a fixed plain bearing (Fig. 268). Let the shaft be made of a steel whose linear expansion coefficient is α_1, and the bearing body be made of an alloy with α_2. The working temperatures are respectively t_1 and t_2.

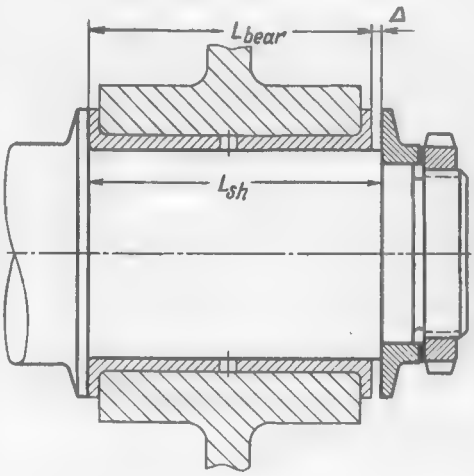


Fig. 268. Determination of thermal end face clearance in a locating plain bearing

When assembled the end face clearance (cold) is

$$\Delta = L_{sh} - L_b$$

where L_{sh} and L_b are respectively lengths of shaft journal and bearing.

When heated to the working temperature t_1, the length of the shaft journal will be

$$L'_{sh} = L_{sh}[1 + \alpha_1 (t_1 - t_0)]$$

length of bearing

$$L_b' = L_b [1 + \alpha_2 (t_2 - t_0)]$$

The end face clearance in the working state

$$\Delta' = L_{sh}' - L_b' = L_{sh} - L_b + L_{sh}\alpha_1 (t_1 - t_0) - L_b\alpha_2 (t_2 - t_0) =$$
$$= \Delta + L_{sh}\left[\alpha_1 (t_1 - t_0) - \frac{L_b}{L_{sh}} \alpha_2 (t_2 - t_0)\right]$$

where t_0 is the assembly temperature.

As the relation $\frac{L_b}{L_{sh}}$ is very close to unity, one may say

$$\Delta' = \Delta + \Delta_t$$

where Δ_t is the thermal change of the clearance.

$$\Delta_t = L_{sh} [\alpha_1 (t_1 - t_0) - \alpha_2 (t_2 - t_0)] \qquad (7.21)$$

Depending on the relationship between α_1, α_2 and t_1, t_2 the original (cold) clearance may increase or decrease. The latter case is dangerous as the shaft end faces may seize.

Let the bearing body be made of aluminium alloy with a linear expansion coefficient $\alpha_2 = 23 \cdot 10^{-6}$, and the shaft be made of a steel with $\alpha_1 = 11 \cdot 10^{-6}$. If we assume 100°C for the working temperature of the bearing body and 50°C for the shaft, the length of the shaft journal 100 mm, the assembly temperature 20°C and the amount of the original cold clearance equal to 0.05 mm, then the thermal change of the clearance according to Eq. (7.21)

$$\Delta_t = 100 [11 \cdot 10^{-6} (50 - 20) - 23 \cdot 10^{-6} (100 - 20)] =$$
$$= 100 (-0.0015) = -0.15 \text{ mm}$$

Hot clearance

$$\Delta' = \Delta + \Delta_t = 0.05 - 0.15 = -0.1 \text{ mm}$$

Thus, in the connection interference of 0.1 mm arises; consequently, the shaft will seize in the bearing. If a minimum clearance is necessary in the working state, e.g., 0.05 mm, then the original cold clearance must equal $0.05 + 0.15 = 0.2$ mm.

The selection of correct face clearances is of utmost importance for multi-supported shafts with bearings spaced at fairly large distances one from another (Fig. 269). Let the front bearing A be the fixed one. To avoid shaft seizure due to the crankcase heating it is necessary to provide adequate clearances between the webs of the crankshaft and the faces of the corresponding bearings. These clearances, designated as Δ_1, Δ_2 and Δ_3, must be proportional to the distances L_1, L_2 and L_3, of the bearings from the fixed one. Using the numerical values of α_1, α_2, t_0, t_1 and t_2 from the previous example

and assuming $L_1 = 300$, $L_2 = 500$ and $L_3 = 700$ mm, we obtain the following values for the thermal changes in the clearances

$$\Delta_1 = 300\,(-0.0015) = -0.45 \text{ mm}$$
$$\Delta_2 = 500\,(-0.0015) = -0.75 \text{ mm}$$
$$\Delta_3 = 700\,(-0.0015) = -1.05 \text{ mm}$$

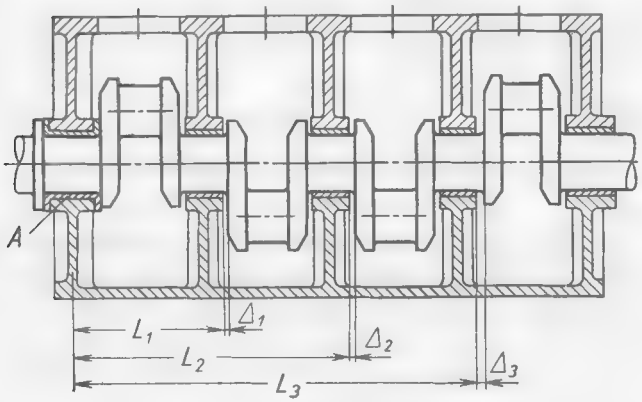


Fig. 269. Determining thermal end face clearances in a multi-support crankshaft design

When fixing the clearances one should add to them the magnitudes of the original cold clearances, which were established with due account to the corresponding shaft and crankcase dimensions.

(*b*) *Position of Locating Bases*

The position of locating bases should be chosen with such consideration that with all possible temperatures, the accuracy of the sizes of the details positioned in the system is not impaired or, if so, to the very smallest degree.

In the bevel gearing unit, accommodated inside a light alloy housing (Fig. 270*a*), the fixing bearing *1* is spaced at distance *L* far from the centre of engaged gears. Elongation of the housing because of heating will make the minor gear shift in the direction shown by the arrow. The major gear will also shift in the same direction but over a smaller distance (this being due to a lesser coefficient of linear expansion of the steel shaft). As a result, the engagement clearance decreases. With certain relationship the gears will start operating under thrust.

In the correct design, illustrated in Fig. 270*b*, the fixing element is rear bearing *2*, spaced at rather a small distance L_1 from the centre

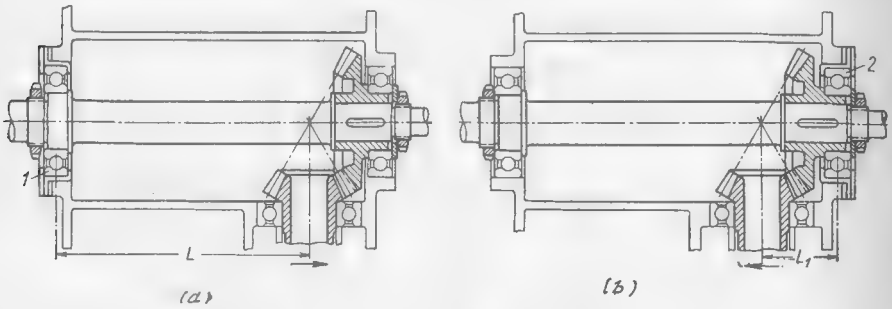


Fig. 270. Fixing a horizontal shaft in a bevel gear unit

of engagement. The displacement of the bevel gears one relative to the other, during heating is much less; in this instance the engagement clearance increases, not decreases as in the previous case.

(*c*) *Assurance of Free Thermal Deformations*

Axial fixing of parts at two points should be avoided as thermal deformations in such cases due to form constraint can cause thermal stresses.

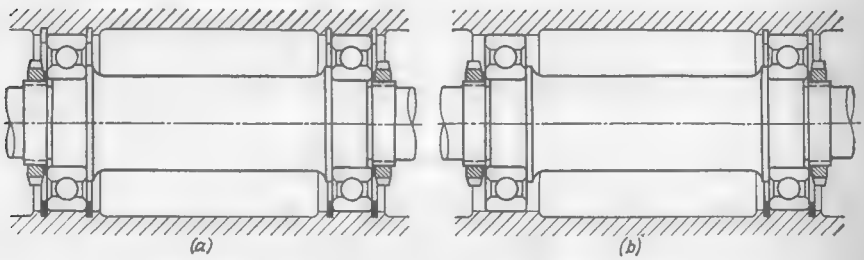


Fig. 271. Fixing a shaft in antifriction bearings

An example of incorrect mounting is the fixing of a shaft in two antifriction bearings simultaneously (Fig. 271*a*). If the bearing body is made of a material whose linear expansion coefficient differs from that of the shaft, and if the shaft and the casing have different working temperatures, then either clearance or interference occur in the unit, causing bearing seizure. Inevitable errors in the axial dimensions of connections during manufacture may, in turn, cause clearance or interference.

Hence, the shaft must be fixed at one bearing (Fig. 271*b*). The second bearing must float, i.e., have free movement in the axial direction.

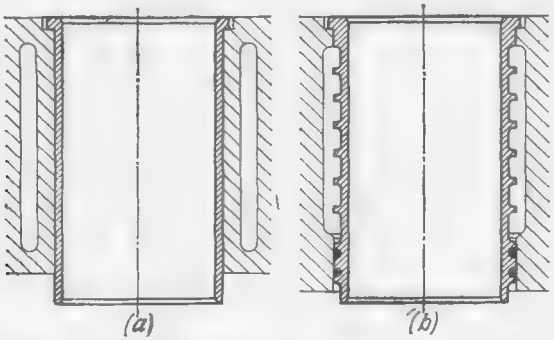


Fig. 272. Mounting a cylinder liner in an internal combustion engine

Another example is presented in Fig. 272, which illustrates a cylinder liner of an internal combustion engine directly being cooled by water. The version, shown in Fig. 272*a*, in which the liner is

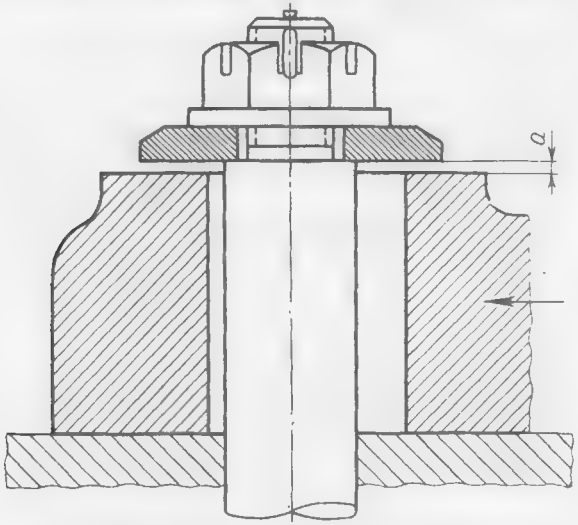


Fig. 273. A turbine housing mounting foot

fixed in two points, by the top shoulder and the sealing shoulder is a mistake. Thermal forces arise in the unit, the liner is heated, compressing the liner and stretching the jacket. In the correct design, pictured in Fig. 272*b*, the liner is fixed by its top shoulder only and the seal can float, therefore the liner freely moves relative to the jacket.

Free movement of the fastening feet of machines and units which heat during operation, must also be assured: one of the feet (or a number arranged in a line) may be rigidly secured and the rest allowed free movement.

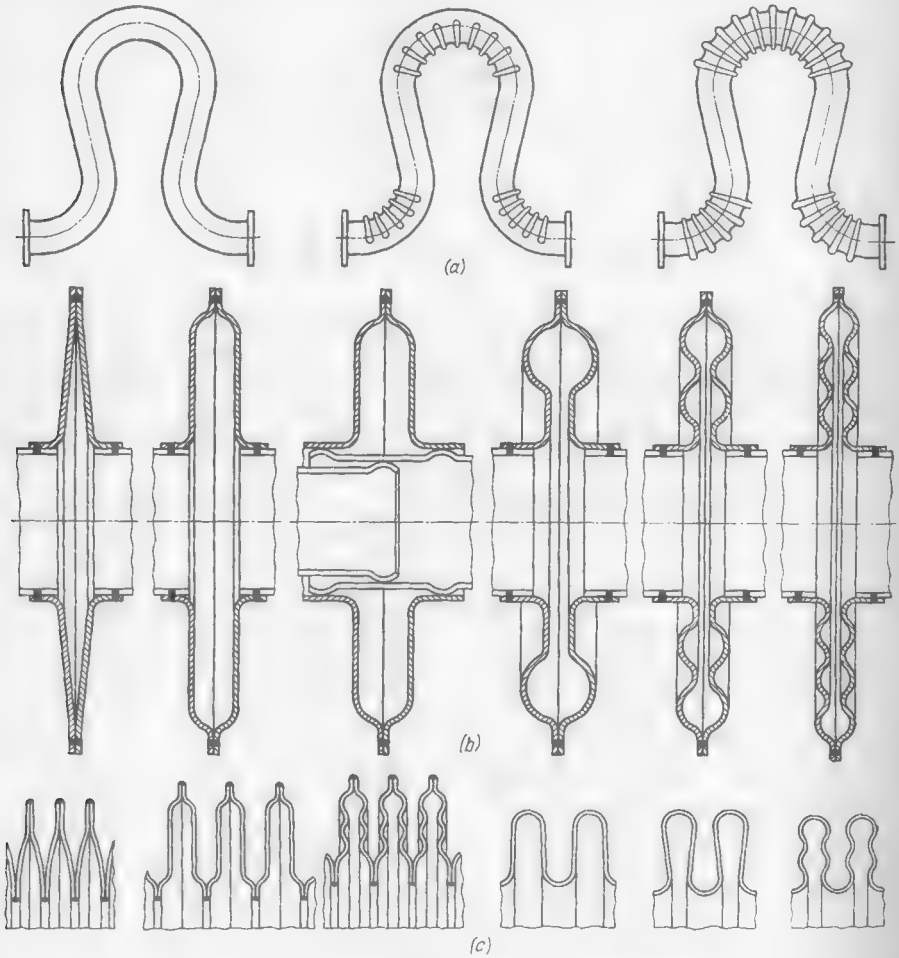


Fig. 274. Thermal expansion compensators

Figure 273 depicts a typical fastening foot design, which fastens a turbine housing to a foundation (the direction of thermal expansion of the housing is shown by an arrow). The foot is secured in place by an anchor bolt, extending through an oblong hole. A clearance $a = 0.05\text{-}0.1$ mm is provided between the bolt washer and the foot end face.

In connections of pipes, conveying not fluids or gases, it is necessary to preliminary supply thermal expansion compensators preventing the onset of thermal forces and pipe deformation.

"Lyre"—shaped compensators (Fig. 274*a*) have large sizes. More compact are the cap-type (Fig. 274*b*) or, even better, bellows-type compensators (Fig. 274*c*).

(*d*) *Positional Changes During Heating*

When designing connections working at high temperatures, it is absolutely necessary to calculate the probable changes in dimensions and relative position of the heated parts.

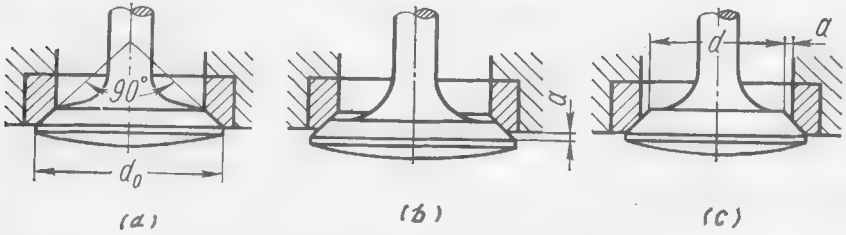


Fig. 275. Position of a valve in its seat

As an example consider the fit in a seat of a discharge valve in an internal combustion engine (Fig. 275*a*).

After being heated, the diameter of the valve head increases by the value

$$\Delta \approx d_0 \alpha_v (t_v - t_0) \tag{7.22}$$

and the valve seat diameter by

$$\Delta' \approx d_0 \alpha_s (t_s - t_0) \tag{7.23}$$

where d_0 = diameter of valve head

α_v = corresponding coefficients of linear expansion of materials of valve and seat

and α_s

t_v = corresponding working temperatures of valve head and seat

and t_s

t_0 = initial temperature (temperature of assembly)

Since the working temperature of the valve head is well above that of the seat, the valve during heating will move out of the seat (Fig. 275*b*) by an amount

$$a = 0.5 (\Delta - \Delta') \tan \alpha/2$$

where α is the included valve face angle.

When $\alpha = 90°$ $a = 0.5 (\Delta - \Delta')$.

Taking into account Eq. (7.22) and Eq. (7.23)

$$a = 0.5 d_0 [\alpha_v (t_v - t_0) - \alpha_s (t_s - t_0)] \tag{7.24}$$

In high-energy engines the discharge valves and seats are made of Cr-Ni austenitic steels (e.g., X13H7C2), whose coefficient of linear expansion at temperatures up to 800°C is equal to $\alpha = (18\text{-}20)10^{-6}$. Assuming the working temperature of the valve head equal to $t_v = 700°C$, and that of the seat $t_s = 300°C$, and the temperature of assembly $t_0 = 20°C$, we obtain

$$a = d_0 0.5 \cdot 20 \cdot 10^{-6} \times$$
$$\times \ (680° - 280°) = 0.004 d_0$$

With a valve head diameter $d_0 = 60$ mm

$$a = 0.004 \cdot 60 = 0.24 \text{ mm}$$

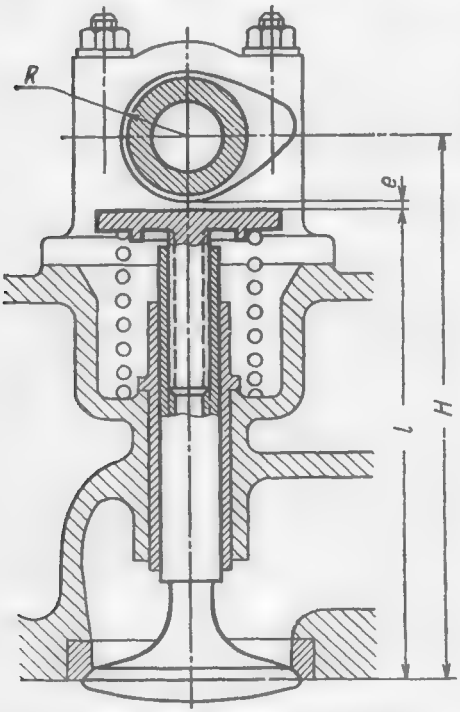


Fig. 276. Valve drive mechanism

To ensure a correct fit of the valve in the seat it is necessary to reduce the minor diameter d of the head (Fig. 275*c*) by the value $2a \approx 0.5$ mm.

Now let us analyse how thermal strains affect the geometry of the valve drive unit. In the simplest version, presented in Fig. 276, the valve is actuated by a cam shaft, mounted in bearings in the engine head (overhead valve drive) and acts directly upon the valve plate.

The clearance between the cam rear surface and the valve plate in cold state

$$e = H - R - l \tag{7.25}$$

In the hot state

$$e' = H [1 + \alpha_h (t_h - t_0)] - R [1 + \alpha_{sh} (t_{sh} - t_0)] -$$
$$- l [1 + \alpha_v (t_v - t_0)] + a = e + H \alpha_h (t_h - t_0) -$$
$$- R \alpha_{sh} (t_{sh} - t_0) - l \alpha_v (t_v - t_0) + a \tag{7.26}$$

where α_h, α_{sh}, α_v = coefficients of linear expansion of the engine head, cam shaft and valve, respectively
t_{sh}, t_h and t_v = respective average temperatures
u = displacement of valve in seat as a result of valve head expansion [see Eq. (7.24)]

Given: $\alpha_h = 11 \cdot 10^{-6}$ (cast iron); $\alpha_{sh} = 11 \cdot 10^{-6}$ (structural steel); $\alpha_v = 20 \cdot 10^{-6}$ (austenitic steel); $t_h = 100°C$; $t_{sh} = 50°C$; $t_v = 450°C$; $H = 150$ mm; $R = 20$ mm; $l = 130$ mm and $a = 0.24$ mm. Clearance variation according to Eq. (7.25) and Eq. (7.26) will be

$$e' - e = 150 \cdot 11 \cdot 10^{-6}(100° - 20°) - 20 \cdot 11 \cdot 10^{-6}(50° - 20°) - \\ - 130 \cdot 20 \cdot 10^{-6}(450° - 20°) + 0.24 \approx -0.7 \text{ mm}$$

To avoid any disturbance of gas distribution phases during starting period the cold clearance in the above case must be made equal to

$$e'' = 0.7 + e_0$$

where e_0 is the warranted clearance.

In other valve drive designs, for instance, with a bottom distribution or with the use of tappets, conrods, levers or rockers, the magnitudes of clearance variation may be still greater and can be calculated in the similar way.

In the latest engines designs, use is made of automatic compensators, which enable clearance in the valve distribution to be maintained nearly constant, regardless of the engine thermal conditions.

(e) *Shape Corrections of Parts*

In a number of cases uneven heating distorts the shape of parts. In such instances the initial form of the part is corrected to calculations which assure that the part takes the required shape as it is heated to operational temperatures.

Similar methods are applied to the pistons of internal combustion engines. The piston has its maximum temperature at the head (Fig. 277*a*). Towards the skirt the temperature falls owing to the removal of heat by piston rings, which transfer heat to the cylinder walls, and also due to the cooling effect of oil splashed from the case onto the internal walls of the piston. When heated, the piston will acquire approximately a conical shape (shown by dashes in Fig. 277*a*). To prevent piston head seizure the piston is machined to an inverted conical form, converging to the head (Fig. 277*b*).

The magnitude of the "cold" clearance between the piston and cylinder walls and the amount of required convergence of the piston head can be found from the following relationship.

Diametral clearance between the piston and cylinder walls in cold state

$$\Delta = D - d$$

where D and d

are the nominal diameters of cylinder and piston, respectively.

In the working state the clearance is

$$\Delta' = \Delta - D\left[\alpha_p (t_p - t_0) - \frac{d}{D}\ \alpha_c (t_c - t_0)\right] \approx$$

$$\approx \Delta - D[\alpha_p (t_p - t_0) - \alpha_c (t_c - t_0)] \quad (7.27)$$

where α_p

and α_c = coefficients of linear expansion of materials of piston and cylinder, respectively

t_p

and t_c = average temperatures of piston and cylinder, respectively

Let the diameter of cylinder $D = 100$ mm, $\alpha_p = 23 \cdot 10^{-6}$ (aluminium alloy), $\alpha_c = 11 \cdot 10^{-6}$ (steel), temperature of cylinder walls

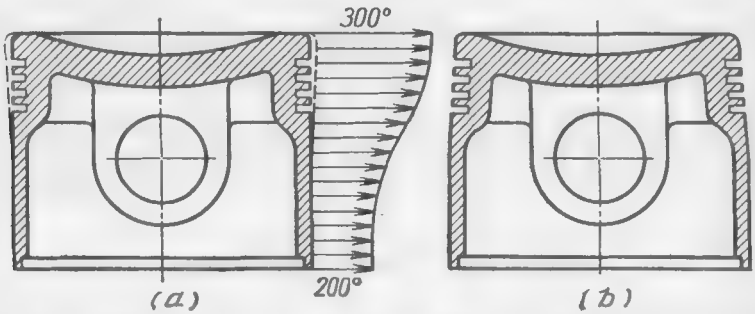


Fig. 277. Piston shape correction to allow for thermal deformation in heating

100°C (liquid-cooled engine), temperature of piston top collar 300°C and bottom collar 200°C.

To assure that the piston will acquire a cylindrical form when heated, it is necessary to assure that in the cold state the diameter of the piston head is less than that below lower piston ring groove by the value

$$\Delta d = 100 \cdot 23 \cdot 10^{-6} (300° - 200°) = 0.23 \text{ mm}$$

The alteration in clearance between the piston and cylinder in the hot state according to Eq. (7.27) is equal to

$$\Delta - \Delta' = 100 \,|\, [23 \cdot 10^{-6} (200° - 20°) - 11 \cdot 10^{-6} \times$$

$$\times\ (100° - 20°)] = 0.32 \text{ mm}$$

Now, if the minimum clearance between the piston and cylinder in the hot state should be equal to, say, 0.3 mm, then the cold state clearance must be equal at the head to $0.3 + 0.32 + 0.23 = 0.85$ mm and below the bottom ring groove $0.3 + 0.32 = 0.62$ mm.

Now determine the design clearance between the rear surface of piston rings and the bottom of piston grooves (Fig. 278).

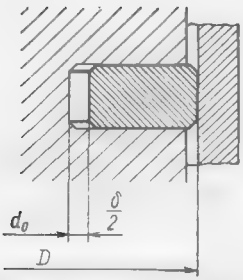


Fig. 278. Determining the clearance between the rear surface of a piston ring and piston groove bottom

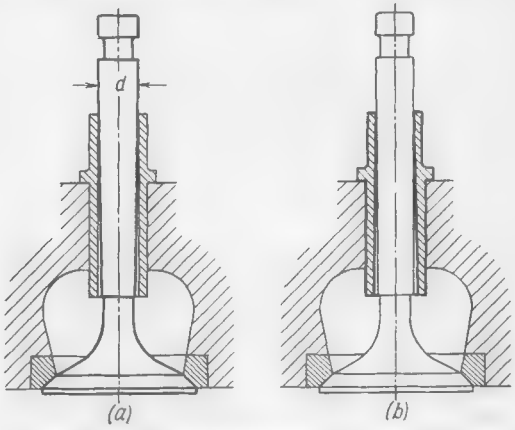


Fig. 279. Shape correction of a valve and guide

After the piston has been heated to the working temperature, the diameter d_0 of the piston groove bottom increases by

$$\Delta d_0 = d_0 \alpha_p (t_p - t_0)$$

and the diameter of the cylinder

$$\Delta D = D \alpha_c (t_c - t_0)$$

If we neglect the change in ring thickness during heating, then the change in diametral clearance between the ring rear surface and inside of the piston groove

$$\Delta\delta = \Delta d_0 - \Delta D = d_0 \alpha_p (t_p - t_0) - D \alpha_c (t_c - t_0) =$$

$$= D \left[\frac{d_0}{D} \alpha_p (t_p - t_0) - \alpha_c (t_c - t_0) \right]$$

Assuming $d_0/D = 0.85$ and substituting numerical values, we obtain

$$\Delta\delta = 100 \left[0.85 \cdot 23 \cdot 10^{-6} (300° - 20°) - 11 \cdot 10^{-6} (100° - 20°) \right] = 0.46 \text{ mm}$$

Let the clearance necessary to meet the requirements of ring normal functioning in working state be equal to 1 mm. Hence, the design (cold) clearance must be equal to 1.46 mm.

Another example of correcting shape is the tapering of exhaust valve stems of internal combustion engines (Fig. 279*a*). Since the operating temperature of the rod upper end is lower than that at the neck (i.e., where the rod joins the head), the diameter of the rod upper end must be (to maintain a constant clearance throughout the guide bush) larger than the diameter of rod near the neck by the value

$$\delta = d\alpha_v \Delta_t$$

where d = nominal diameter of rod
α_v = coefficient of linear expansion of valve material
Δt = difference in temperature between the neck and upper end of the rod

For a valve, made from austenitic steel ($\alpha_v = 20 \cdot 10^{-6}$), rod diameter $d = 12$ mm and $\Delta t = 200°C$

$$\delta = 12 \cdot 20 \cdot 10^{-6} \cdot 200 \approx 0.05 \text{ mm}$$

Correction can also be accomplished by tapering the valve guide bore making it larger at the valve head end (Fig. 279*b*).

7.3. Temperature-Independent Centering

The usual methods of centering cylindrical surfaces are unsuitable when substantial thermal strains occur in the connecting parts.

If a female part has a higher temperature or is made of a material with a higher thermal expansion coefficient than the male part, then clearance will occur in connection, thereby disturbing the accuracy of centering. If, however, the situation is opposite to that described above, then interference occurs which overloads the connection and deforms the mated parts, which finally also disturbs centering.

This should especially be considered when designing heat engines (e.g., gas turbines) with large-diameter housings, which are often made of different materials.

Assume that the circular housing members of an axial compressor and turbine are connected and centred relative to each other by a shoulder to the class A_{2a}/C_{2a} fit; one of the members is made of a light alloy with a coefficient of linear expansion $\alpha_1 = 23 \cdot 10^{-6}$; the other of steel with $\alpha_2 = 11 \cdot 10^{-6}$. The diameter of the centering shoulder $D_0 = 1000$ mm.

A hole, made to 2a class accuracy may deviate within the limits of 0 to 0.13 mm, and the shoulder diameter from 0 to —0.09 mm. Consequently, the connection, assembled cold may have clearance from 0 to 0.22 mm.

Let the operating temperature of the housings be 150°C. Then the diametral expansion difference of the centering surfaces

$$\Delta D = D_0 t_{op} (\alpha_1 - \alpha_2)$$

Substituting numerical values we shall obtain

$$\Delta D = 1000 \cdot 150\,(23 - 11) \cdot 10^{-6} = 1.8 \text{ mm}$$

Adding this value to the cold clearance value (0-0.22 mm), we obtain the hot-state clearance: minimum 1.8 mm and maximum 2.02 mm. Naturally, the centering accuracy is completely lost.

Fig. 280 presents the designs of flanges, made of metals with different coefficients of thermal expansion. Shown also are some techniques, by which centering can be accomplished in the presence of thermal strains.

In the designs shown in Fig. 280*a* steel flange *1* is centred by the shoulder of the housing component *2* made from aluminium alloy.

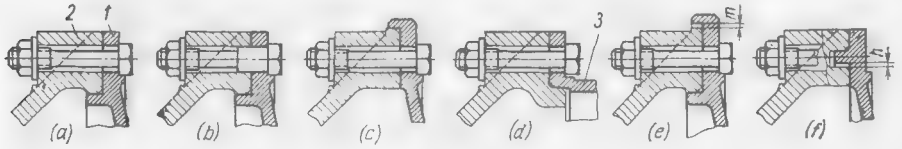


Fig. 280. Methods of centering flanges made from materials with different linear expansion coefficients

After the system is heated clearance appears in the joint. Centering, in effect, is implemented by an indefinite action of fastening bolts.

More positive centring is effected when the joint is tightened by fitting bolts (Fig. 280*b*). However, with heating interference occurs at the joint deforming the unit.

Interference also occurs when centering by an external shoulder on the steel flange (Fig. 280*c*).

The above-described methods of centering are applicable when one of the connecting parts can yield in the radial direction, for instance, if the steel flange is part of a thin-walled cylindrical housing *3* (Fig. 280*d*), which may somewhat expand radially. In this case the stresses in the unit decrease.

Occasionally a double centering system s applied (Fig. 280*e*). In the cold state the connection centres against the internal shoulder of the steel flange. The external shoulder is made with clearance m equal to the difference of the thermally increased diameters of the aluminium and steel flanges. When heated, the centering function is implemented by the external shoulder and clearance occurs at the internal shoulder. In the heating period the centering becomes uncertain between the extreme limits of temperature.

An alternative of this technique is the centering by means of a shoulder which enters with an internal clearance h into a circular recess of the mating part (Fig. 280*e*).

(*a*) *Star-Radial Centering*

In cylindrical parts subjected to uniform thermal expansion all elements will displace along radii, converging on the axis of symmetry. If the centering elements are arranged along radial lines, the centering will be maintained under any thermal deformations in the system. The number of centering elements must not be less than three.

Such a type of centre alignment is termed star-radial.

Examples of star-radial centering are pictured in Fig. 281 (the female part is made from aluminium alloy, and the male from steel).

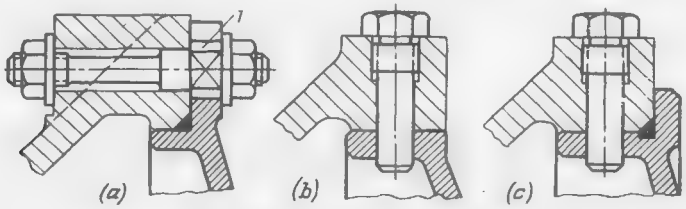


Fig. 281. Star-radial centering

In the version, Fig. 281*a*, the centering elements are bolt necks *1* with flats. The necks tightly fit in radial slots in the flange. The flange is tightened to the flange with such a force that the friction force in the joint is less than the thermal forces arising during heating or cooling. Occasionally the system can be tightened until the nut thrusts against the bolt neck in order to obtain a minimum axial clearance (some hundredths of a millimetre) in the connection. Centring by a shoulder is not necessary in this case (the shoulder shown in Fig. 281*a* serves only to accommodate packing).

A version of the star-radial method is pin (stud) centering. The centering pins are tightly fitted into holes which are machined on assembly in the mating part (Fig. 281*b*). The shoulder in this case is only a preliminary centering flange for the machining.

This technique does not implement axial tightening of parts; the pins only fix the parts in the axial direction. The sealing of the joint can be attained by introducing elastic packing elements into the joint (Fig. 281*c*).

Figure 282*a-i* shows a number of star-radial centering techniques applied to parts transmitting torque. Centering is accomplished by means of keys—prismatic (Fig. 282*a*,*b*) or circular (Fig. 282*c*), bolts with flats (Fig. 282*d*), face cams (Fig. 282*e*), splines (Fig. 282*f*) and radial pins (Fig. 282*g*, *h*).

Preliminary centering of connecting flanges by a cylindrical surface (Fig. 282*g*) is used whenever a female part expands under working temperatures more than its male counterpart. In the opposite case radial clearance is provided between the male and female parts

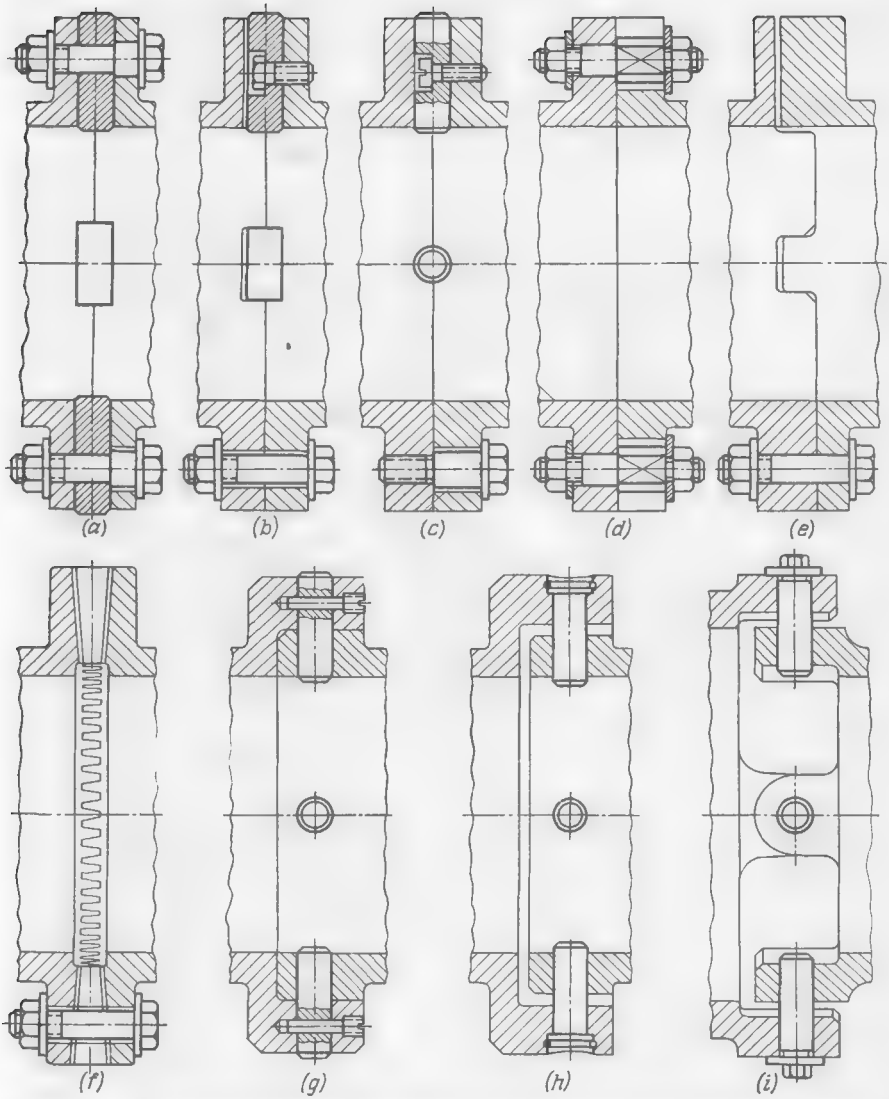


Fig. 282. Star-radial centering of flange connections transmitting torque

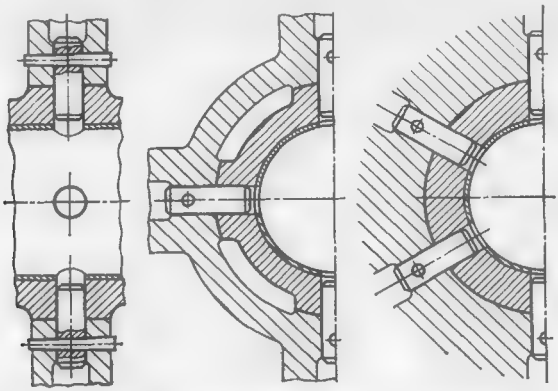


Fig. 283. Star centering of plain bearings

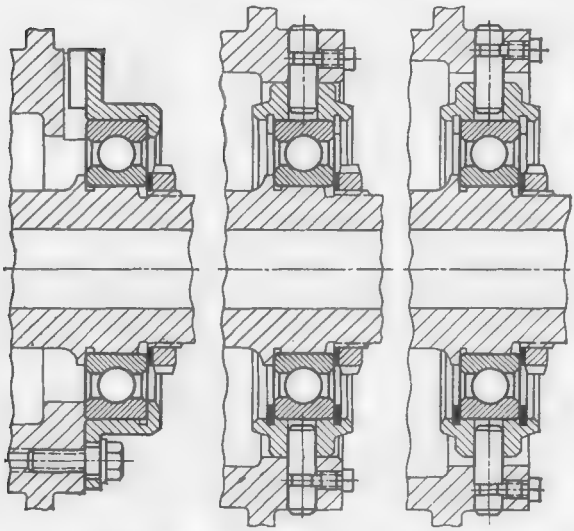


Fig. 284. Star centering of antifriction bearings

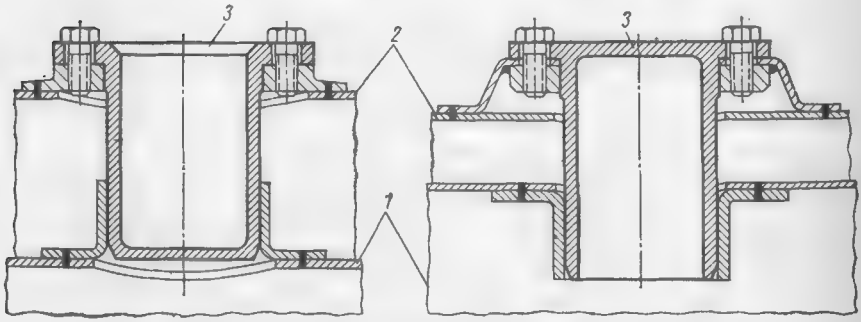


Fig. 285. Star-type suspension of combustion chambers

(Fig. 282*h*, *i*). Machining of holes intended to receive centering pins is done separately (to a jig) or in assembly using dummy centring rings.

Figures 283 and 284 illustrate star-radial centering of plain bearings and antifriction bearings whose housings are made of light alloys.

A star suspension is often applied when parts operate under high temperatures and temperature gradients, for example, in furnaces. Figure 285 illustrates a suspension design of aircraft turbine engine combustion chambers *1* inside casings *2* by means of radial centering sleeves *3*.

Figure 286 suggests an alternative of star-radial suspension of fire-box *1* on longitudinal ribs *2* which assure freedom for radial and axial strains.

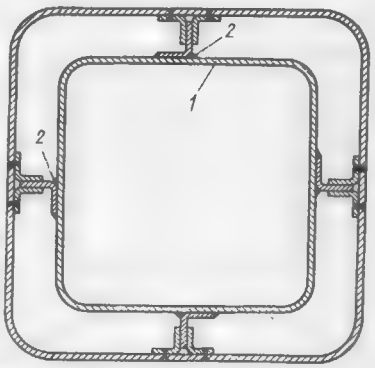


Fig. 286. Star-axial suspension of a fire box

(*b*) *Alignment of Fitted-On Parts*

The problem of temperature-independent alignment of parts is encountered when fitting turbine rotors on shafts of centrifugal and axial compressors, etc. If the rotor temperature is too high (turbine runners) or if it is made from some light alloy (centrifugal and axial compressors), then clearance will appear in the fitting zone leading to disbalance and runout. In high-speed rotors the clearance increases still more under the action of tensile centrifugal forces which reach their maximum near the rotor bore. Under such circumstances it is necessary to paralyze the effect of thermal strains and hub tension.

An effective means is the cooling of rotors. This technique is widely used in gas turbines. Cooling air, taken from compressor first stages, flows around the runners and then enters the turbine duct. Cooling of steam turbine rotors is more difficult.

Examples of temperature-independent centering of fitted parts are illustrated in Fig. 287.

With double centering (Fig. 287*a*) the cold rotor is centered by its bore upon the shaft. In the working state, when the diameter of the hub increases, centering is accomplished by means of ring shoulders, which embrace the hub from both sides. In the interval between the extreme positions the rotor is decentralized and this can cause harmful vibrations.

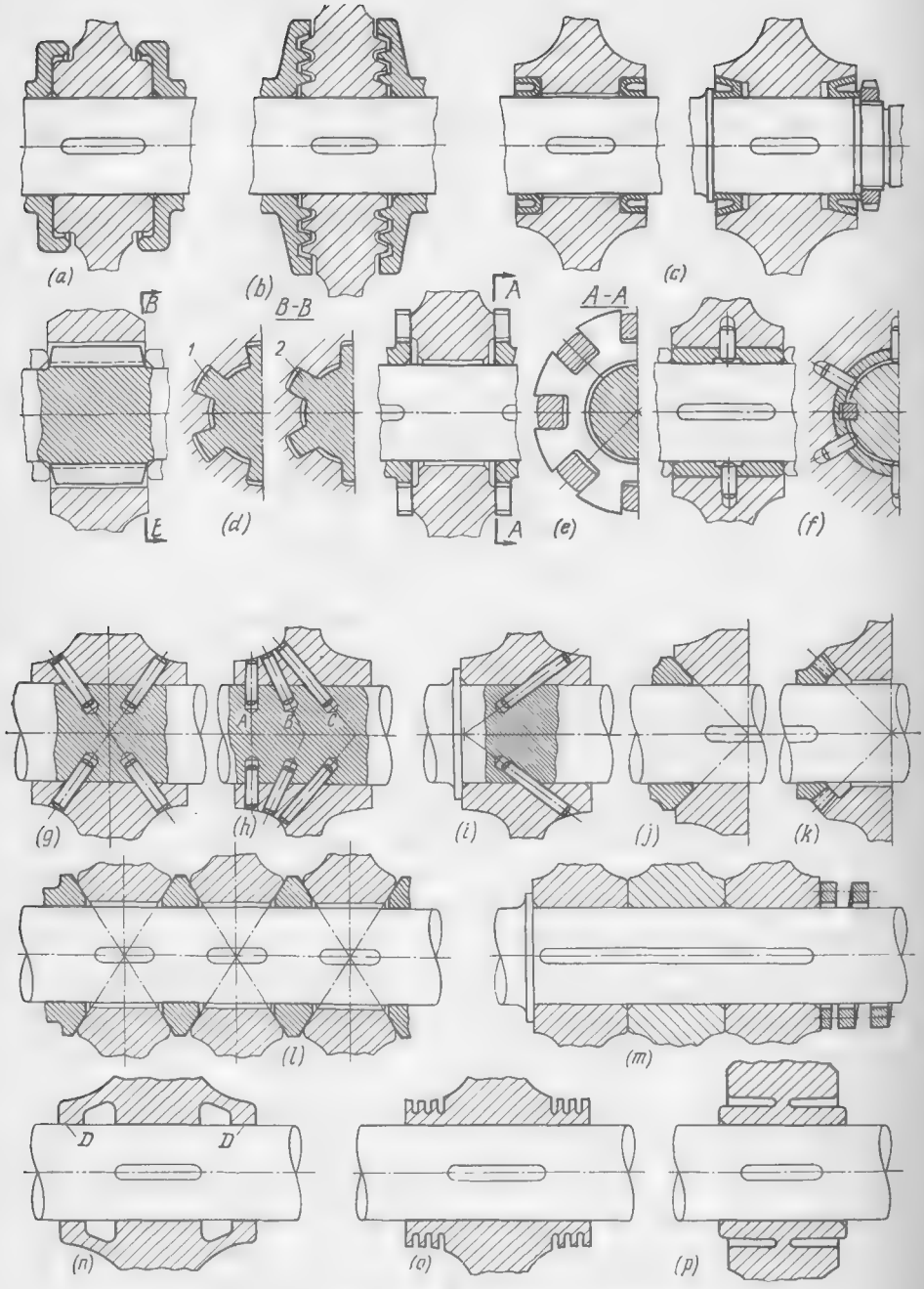


Fig. 287. Centering of fitted-on components

Multi-stepped centering is accomplished by comb-shaped disks with clearances successively increasing towards the periphery (Fig. 287*b*). As the hub is heated its dimensions increase and new ridges come into action, thus the alignment is kept at all stages of heating.

Sometimes use is made of spring C-shaped rings (Fig. 287*c*) fitted between the shaft and hub. In this case the possible rotor displacements relative to the shaft are confined within the elastic strain limits of the rings. As a result radial and axial runout of the rotor may occur.

Star centring is effected by side faces of splines *1*, whose planes converge at the shaft axis (Fig. 287*d*). With uniform heating and axisymmetrical hub tension by the centrifugal forces the system retains its geometric similarity, and the centering is kept under any operating conditions.

Practically the same result is obtained with splines *2* centering on their working faces. Deviation from correct centring is less with thinner splines, i.e., the larger their number.

Figure 287*e*, shows a star method where centering is on the faces of teeth entering radial slots of the drive disks.

When parts with long hubs are fitted upon shafts changes in hub axial dimensions should be considered. When heating symmetrically in the equatorial and meridional planes each heated point of the part will move along rays outgoing from the geometrical centre.

The simplest solution is to arrange the centering elements within the part's meridional plane of symmetry along radii, which converge on the axis. This principle underlies the system of pin bushes (Fig. 287*f*) often applied in turbine design practice. The pins are mounted in intermediate bushes since other ways of assembly are impossible. The bush is introduced into the rotor hub, the pins are fitted from inside and the rotor then assembled onto the shaft.

The bush is interference fitted upon the shaft (sometimes by tightening on a conical surface). Since the bush is loaded with inappreciable centrifugal forces, its actual dimensions remain constant and the interference fit is preserved. The resultant system provides freedom of thermal deformation in the radial and axial directions (on each side of the pin location plane).

The manufacture of joints presents considerable difficulty. First, the holes in the hub are drilled and reamed with the aid of special heads in which the cutting tools are arranged at the right angle to the axis. Secondly, the holes in the bush and the rotor must be fully concentric and coincident with each other.

Correct star-like alignment can be accomplished in another, still more practical way—by introducing pins from outside into the holes, which have been machined in assembly. To preserve the centring and avoid any changes in the position of the rotor meridional

symmetrical plane the hole axes must converge onto the shaft axis in this plane (Fig. 287*g*). The same effect is attained when pins are arranged in a row (to the left or to the right from the rotor plane of symmetry).

Nevertheless, the system of oblique pins fails to assure true alignment if the hub changes its dimensions when affected by tensile forces. The centrifugal forces acting normally to the shaft axis will bend the pins. Hence, the system described above is applicable only in those cases when thermal deformations predominate, and tensile ones are small. The closer the pins to the axis of symmetry of the part, the better the centering accuracy in the presence of centrifugal forces.

Correct alignment can be ensured also under tensile stresses, if the pins are arranged radially and displaced relative to the axis of symmetry (position *A*, Fig. 287*h*). However, in this case the axial thermal deformations will be directed away from the plane of the pins and the meridional symmetrical plane of the rotor will, when affected by thermal deformations, shift along the shaft.

The rotor plane, which does not change its position relative to the shaft, is determined by the position of points, where the axes of the pins intersect the shaft axis (positions *A*, *B* and *C*).

If a part to be aligned has a butting face, which fixes the direction of axial deformations and if axial thermal deformations predominate (the case of long hubs), then the pin axes must converge in the thrust shoulder plane (Fig. 287*i*). This provides free thermal expansion of the hub.

Another form of star centering is mounting the rotor on cones whose generatrices converge in the meridional symmetry plane of the rotor (Fig. 287*j*). In this case the requirements for correct centering during thermal movements and for a constant position of the meridional symmetry plane are fully met. Torque can be imparted to the rotor through the agency of a key, splines or bevel teeth (Fig. 287*k*).The system will not assure centering if, under the action of tensile forces, the dimensions of the bore increase. An exception is the case when the cones are tightened by a spring, which continuously eliminates clearance on the fitting surfaces. The cone angle must be less than friction angle (to return the hub to its initial state after cooling).

Should several rotors be set in series (Fig. 287*l*), the cones ensure correct radial alignment and keep the position of the meridional symmetry planes of each rotor constant on the shaft, and also prevent axial compressive stresses in the hub and tensile stresses in the shaft during temperature variations.

Occasionally spring tightening (Fig. 287*m*) is applied, which softens axially directed stresses in the system, but does not assure radial alignment of the rotor or its constant axial position on the shaft. In this instance the symmetry plane of the rotors during

thermal deformations shifts by an amount proportional to their distance from the fixed shoulder.

Correct alignment can also be accomplished by removing the centering collars from the zone of active tensile stresses. With this aim the centering surfaces D are separated from the rotor body by annular recesses (Fig. 287*n*). Thus, the centering collars are practically relieved from tensile stresses and keep their original dimensions and fit on the shaft. If the transition from the rotor body to the centring collars has a certain form, the fit can become even tighter as a result of rotor body tension which closes and compresses the fitting collars.

The system will assure good alignment during thermal deformations of the rotor if the heat transfer from the rotor body to the fitting collars is reduced by decreasing the cross sections of the transient areas and if the fitting collars are simultaneously cooled by fins (Fig. 287*o*).

A unique design is presented in Fig. 287*p*. Here the rotor hub is nearly cut into two parts by deep annular grooves: a solid part calculated to take up centrifugal and thermal forces, and a thin-walled subhub centering bush. The dimensions of the bush isolated from tensile stresses and from heat transfer from the rotor remain practically unchanged, thus providing correct rotor alignment under any conditions. The design is used in stationary installations. In aircraft turbines and compressors the gyration forces arising in the course of aircraft manoeuvres may cause overstresses in the connecting web.

7.4. Heat Removal

An active means of lowering thermal stresses and deformations, of avoiding buckling and of keeping material strength is the reduction of temperature and temperature gradient. This can be realized by isolating the component from the source of heat or by increasing thermal transfer into the ambient medium. With particularly high temperatures it is good practice to introduce cooling systems with forced delivery of cooling agent (air, oil, water, etc.).

The design of a disk friction clutch, having one friction lining attached to the clutch body and the other, to the pressure disk (Fig. 288*a*) is nonrational because the heat, emanating during engagement, flows into the thin disk and overheats it. Much more advantageous is the design where the friction linings are attached to the clutch disk (Fig. 288*b*). Due to their heat-insulating properties the linings reliably protect the thin disk against overheating; the heat, evolved during engagement, passes to the solid clutch housing and pressure disk, which due to their heat capacity are heated only slightly.

Heat transfer can be intensified by eliminating thermal resistances. In a bank-type water-cooled engine with dry liners (Fig. 289*a*)

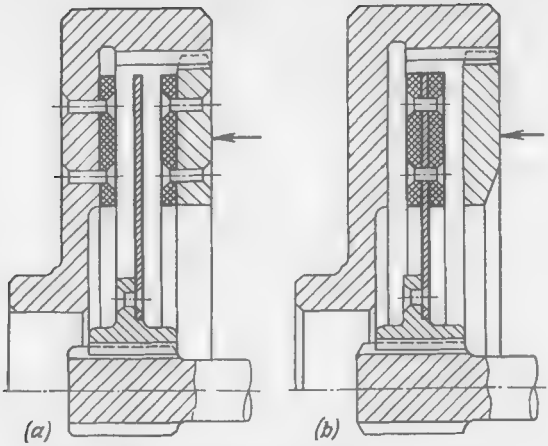


Fig. 288. Fastening of friction linings

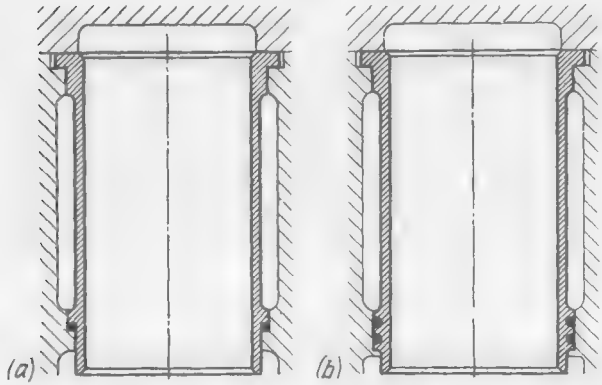


Fig. 289. Increase of heat removal by eliminating superfluous thermal resistances (a liquid-cooled engine cylinder liner)

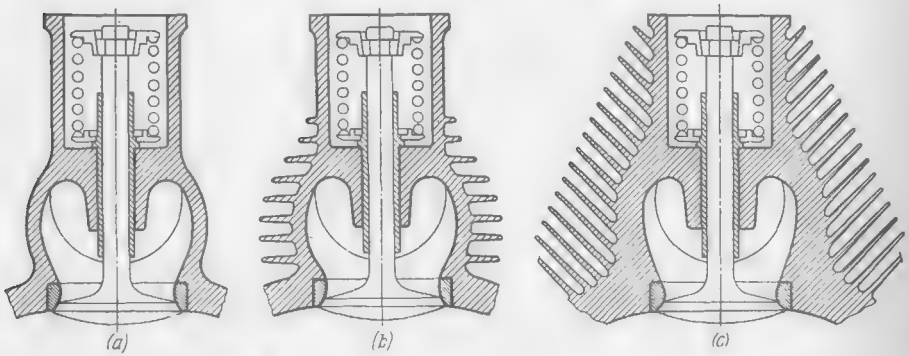


Fig. 290. Exhaust pipe designs in air-cooled engines

heat transfer from the liners into the cooling water is impeded by the superfluous wall, the inevitable oil film and pollutions on the cast surfaces. The temperature of liners directly in contact with circulating water (Fig. 289*b*) is much lower.

Obsolete designs of an air-cooled engine exhaust pipe are presented in Fig. 290*a*,*b*, with one modern design of the same element, having strongly developed ribbing and better heat transfer, pictured in Fig. 290*c*.

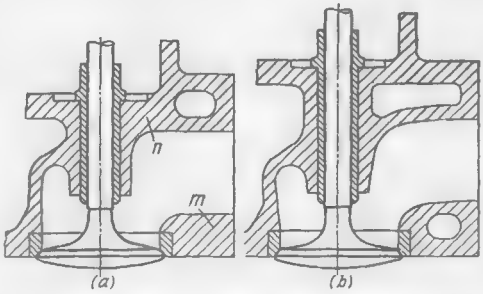


Fig. 291. Cooling of the guide bush and seat of an exhaust valve

Uniform cooling of the seat and guideway areas of the valve is essential, otherwise the seat may lose its cylindrical form, which consequently disturbs valve function. An incorrectly designed exhaust pipe for a water-cooled engine is shown in Fig. 291*a*. The error consists in the single-sided intake of cooling water, which results in poor cooling of solid sections *m* and *n*. In the correct design, shown in Fig. 291*b*, cooling water is circulated around the seat and guideway.

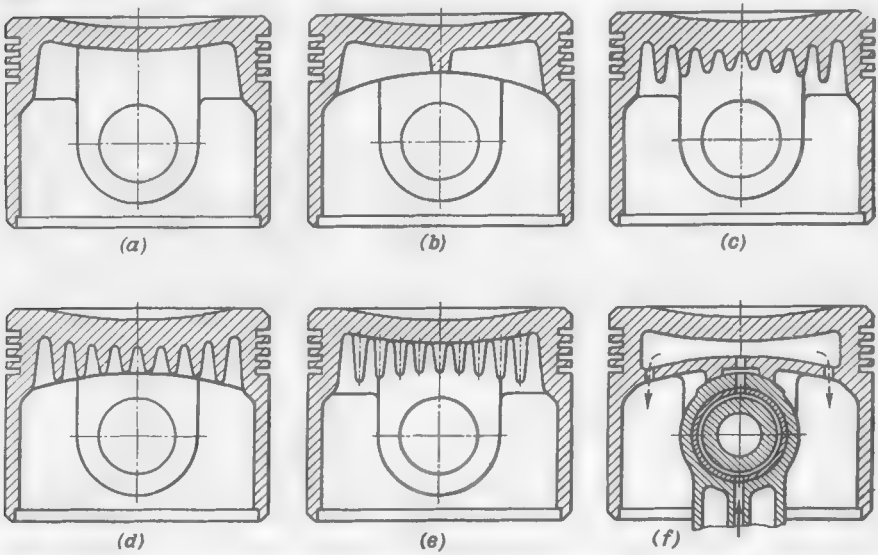


Fig. 292. Increasing heat removal from the piston head in an internal combustion engine

Figure 292, shows methods of intensifying heat flow from internal combustion engine pistons. The piston crown is cooled mostly with oil thrown up from the engine crankcase. To improve heat removal the underside of the crown is ribbed with cruciform (Fig. 292*b*), longitudinal (Fig. 292*c*) or wafer-like (Fig. 292*d*) ribs, which, in addition, increase the strength and rigidity of the crown. The greatest cooling surface in conjunction with the smallest weight is given by acicular coolers (Fig. 292*e*), but they do not add to tl › rigidity of the crown.

In high thermally stressed pistons forced oil cooling is used (Fig. 292*f*). Cooling oil is delivered from the crankshaft journal through bores in the connecting rod. Through a bore in the connecting rod small end oil flows to a cavity under the piston crown and thence back into the crankcase.

(*a*) *Increasing Internal Heat Transfer*

The use of materials with high thermal conductivity facilitates heat transfer from hot areas to cooler ones and lowers temperature gradients.

In components made from materials of low heat conductivity the internal transfer of heat can be assisted by introducing inserts made

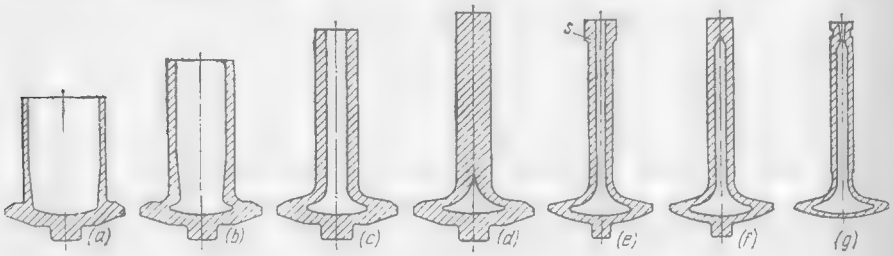


Fig. 293. Successive steps of extruding hollow valve

from metals of high heat conductivity (aluminium, copper) or by filling internal spaces with a liquid heat carrier (e.g., some low-melting metal). The latter method is widely applied in the design of exhaust valves with sodium-cooling. Here the use of a liquid heat-carrier is particularly advantageous because, owing to valve reciprocations, the heat-carrier continuously moves, thus energetically transferring heat from the valve hot head into relatively cooler valve stem.

Metallic sodium possesses a number of valuable properties, which make it a good heat-carrier: low melting temperature (97°C), high heat capacity (0.27 cal/kg °C), low density (0.97 kgf/dm³ in solid

state and 0.74 kgf/dm³ in liquid state). Boiling point is 880°C. An extremely high latent heat of evaporation (1100 cal/kgf) assures good heat absorption potentialities in the event of short, abrupt temperature elevations above 880°C.

Production of hollow valves is difficult. Nevertheless, the higher expenses are well repaid due to greater reliability and longer service terms.

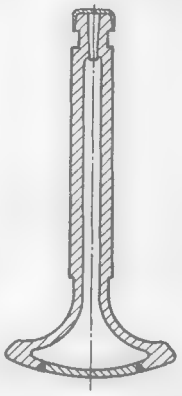


Fig. 294. A hollow valve with welded-on head

The production of hollow valves by extrusion technique begins with drawing a blank in the form of a hollow sleeve (Fig. 293*a*), which is then upset in several passes until the cylindrical portion of the cavity is completely confined by forging (Fig. 293*b-d*). This is followed by drilling and reaming the hole and dressing of external surfaces (Fig. 291*e*). For forging the rod end an allowance *s* is left. After forging the end (Fig. 293*f*), a taper hole is drilled and reamed to suit a sealing plug (Fig. 293*g*). The external surfaces of the valve are preliminarily machined and it is then filled in an inert atmosphere with sodium at 200-300°C to approximately 60% of its volume. The hole is then sealed with a taper plug, and the rod end coated with stellite. Then the valve is finish machined.

Hollow valves can be made simpler by welding the heads on separately (Fig. 294). After welding the valve head spherical surface, the chamfered surfaces and end of the rod are coated with stellite. Then the valve is ground and polished all over.

However, welding can only be applied to some valve steels. The highest heat-resistant steels of martensitic-austenitic grades cannot be welded. Furthermore, welded valves are not as strong as extruded valves.

Strengthening of structures

This section deals with strengthening techniques, which allow certain stresses to be induced in engineering structures that are oppcsite in sign to the working stresses. Two main methods are applied, elastic and plastic.

8.1. Elastic Strengthening

With elastic strengthening the system is preliminary given deformations which are opposite to those of working loads.

A classical example of this type of strengthening is in truss beams (Fig. 295). The system incorporates tensors *1* which are tie-rods made

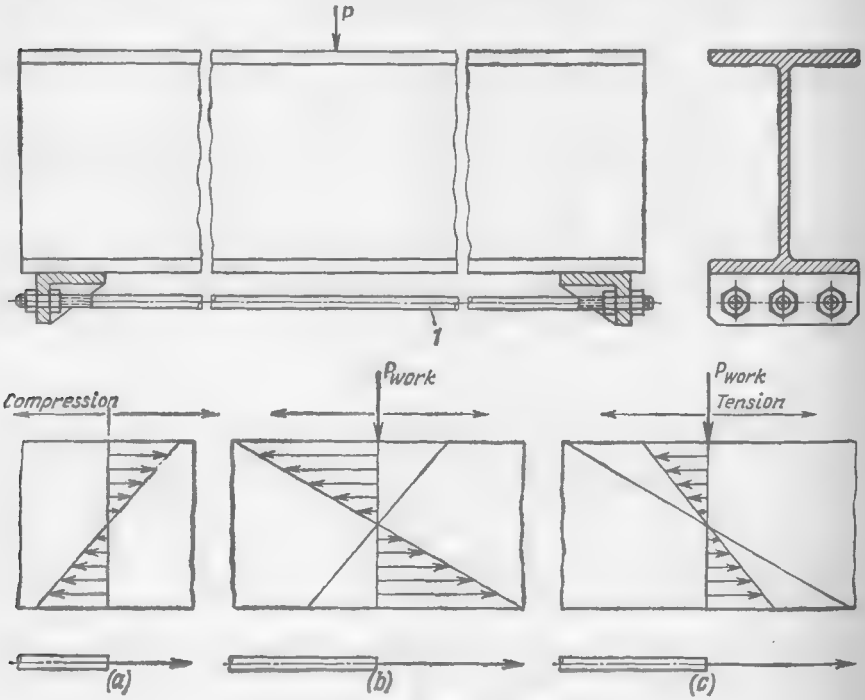


Fig. 295. Truss beam diagrams

of a high-strength material. As these rods are tautened, preliminary stresses are induced in the beam: compressive stresses on the side closer to the rods, and tensile stresses on the opposite side (Fig. 295*a*). Application of working load P_{work} causes stresses of the opposite sign (Fig. 290*b*). Summation of the preliminary and working stresses significantly lowers the final stresses existing in the beam (Fig. 295*c*). Obviously, the tensile stresses in the rods will increase.

Recently the manufacture of prestressed beams has been mastered. Into the flange opposite to the loaded side of the beam are rolled rods, made from high-strength wire, which are prestressed mechanically or thermally (Fig. 296*a*). Such beams can safely be cut into any lengths without impairing the prestressed properties of resultant pieces.

In the other design (Fig. 296*b*) a prestressed strap made from high-strength sheet steel is attached to the lower flange, it is welded to steel beams or riveted to light-alloy ones.

Another example of elastic strengthening is the hooping of containers, made from light alloys, with steel wire (or ribbon), in one or several rows (Fig. 297*a-c*). During coiling compressive stresses are induced in the vessel walls (Fig. 297*d*). Being deducted from the tensile stresses due to internal pressure (Fig. 297*e*), these stresses significantly lessen the resultant stresses in the vessel walls (Fig. 297*f*). Tensile stresses in the wire increase due to the internal pressure.

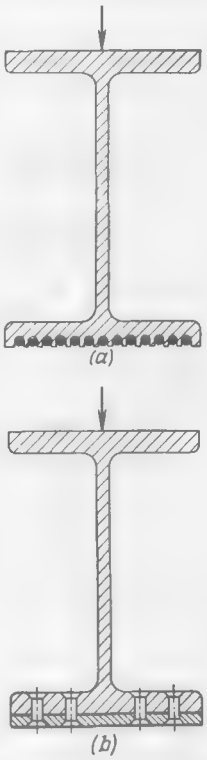


Fig. 296. Prestressed beams

Such systems are practical, however, only if the material of fastening elements is stronger than that of fastened parts. Introduction of preliminary tightening relieves the weaker material and makes the structure, as a whole, stronger.

A variant in elastic strengthening is the hooping of hollow thick-walled cylindrical components subjected to high internal pressures (strengthening of high-pressure vessels, frettage of gun barrels, etc.). In this case it is not obligatory for the fastening elements to be stronger than the fastened ones; the strengthening effect is achieved here owing to the singular distribution of stresses throughout the cross-sections.

According to Lamé, the maximum stresses in a thick-walled vessel subjected to internal pressures, occur at the inside wall and decrease towards the outside (Fig. 298*a*). To attain greater strength the com-

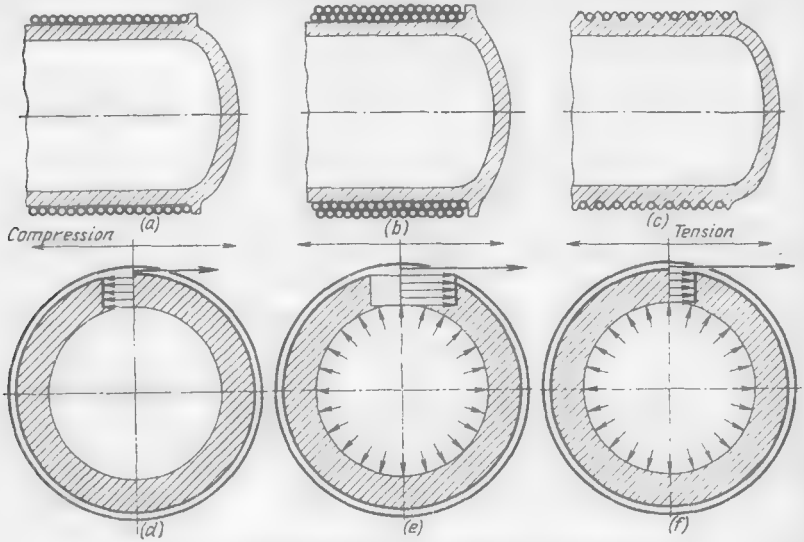


Fig. 297. Strengthening of cylindrical containers

ponent is made of two tubes; the inner tube is press-fitted into the outer one so that high interference is assured. Thus, the outer tube is subjected to tensile stresses, and the inner tube to compressive

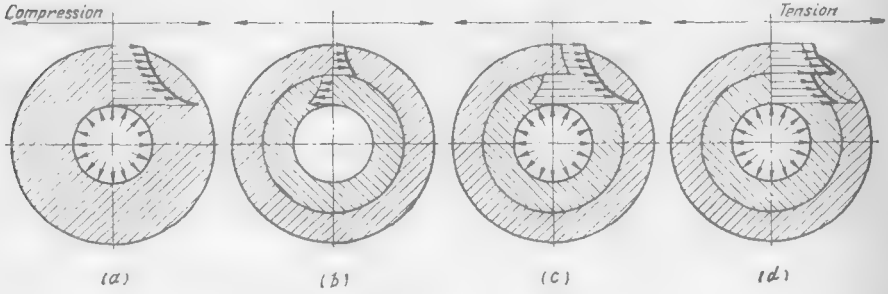


Fig. 298. Hooping of gun barrels

stresses (Fig. 298*b*). As a result of the addition of the preliminary and working stresses (Fig. 298*c*) the tensile-stress peak at the inside wall is lowered (Fig. 298*d*), the stresses throughout the entire cross-section are levelled out and the strength of the system is enhanced.

8.2. Plastic Strengthening

In applying this technique the parts undergoing the greatest loads during operation are subjected to preliminary plastic deformations, thus creating residual stresses which are opposite in sign to the working stresses.

(*a*) *Strengthening by Overloading*

Strengthening by overloading means that the part is subjected to a higher than working load in the same direction which causes plastic deformation in the most heavily stressed areas.

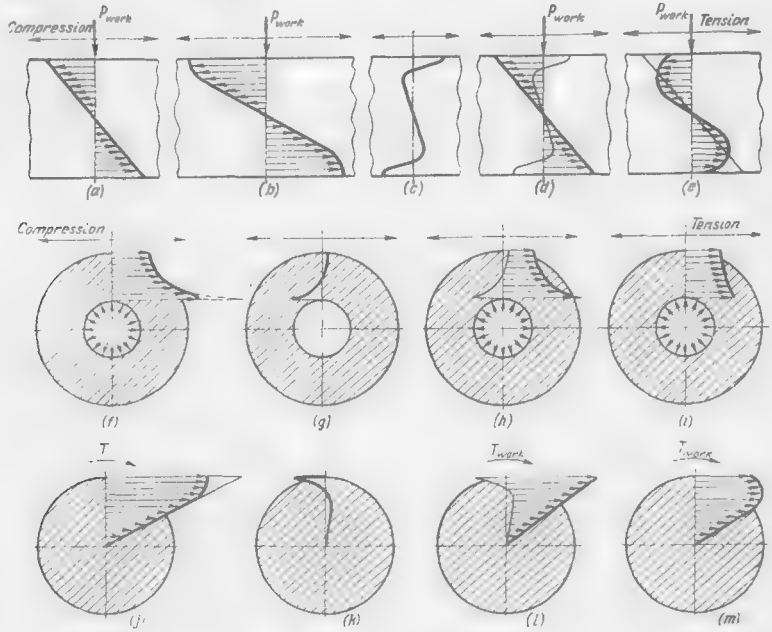


Fig. 299. Strengthening by overloading

When bending a beam by a transverse force P_{work} in the upper fibres of the material arise compressive stresses, and in the lower ones tensile stresses (Fig. 299*a*). Let us now exert upon the beam larger force P, that produces plastic deformation in the extreme fibres (Fig. 299*b*). As a result, the upper fibres are shortened, and the lower ones lengthened. The middle fibres remain in the elastic state. After the strengthening load is removed, the elastic middle while returning into original state stretches the previously compressed upper fibres and compresses the previously stretched lower fibres,

producing in them stresses of opposite sign to the working stresses; and reactive stresses will be induced in the middle (Fig. 299*c*).

Should the beam so stressed be subjected to the action of the work load P_{work} (Fig. 299*d*), then the residual and working stresses add algebraically. The resultant stresses in the extreme fibres are much less (Fig. 299*e*) than the stresses in a beam not subjected to strengthening. Consequently the beam can be loaded with a heavier force, provided the critical limit is not exceeded.

Similarly, thick-walled cylindrical vessels are strengthened by preloading them with higher internal pressure (e.g., autofrettage of artillery gun barrels).

In the vessels pressures are created which cause plastic tensile stresses in the internal wall layers (Fig. 299*f*). After releasing the pressure the elastically stressed basic material of the wall regains its original state compressing the plastically deformed internal layers producing in them residual compressive stresses (Fig. 299*g*). The tensile stresses, which occur in the wall under the action of working pressure (Fig. 299*h*), are partially equalized by the compressive prestresses. The peak of stresses at the internal surface is lowered and the stress distribution throughout the wall becomes more uniform (Fig. 299*i*), making the vessel stronger.

The overloading method is applied also for strengthening torsion bars (e.g., presetting of helical springs). In this case the bar is subjected to a higher torque moment M_{trq}, which induces plastic deformations of shear in the extreme fibres (Fig. 299*j*). After the strengthening load is removed the elastic core is stretched entraining with it the plastically deformed fibres causing in them stresses opposite in sign to the shearing stresses given by the working load (Fig. 299*k*).

Now if some working torque moment $M_{work\ trq}$ is applied to the bar (Fig. 299*l*), the residual stresses add to the working stresses, thus lowering the resultant stresses (Fig. 299*m*).

Strengthening by overloading is applied only to materials possessing sufficient plasticity. In brittle materials overstresses may cause microcracks and flaking in the stretched layers leading to material failure. Similar picture may be observed in plastic materials subjected to high deformation. For these reasons the level of plastic deformation is confined within certain limits, admitting only overstresses which do not exceed 1.1-1.2 of the $\sigma_{0.2}$ yield point.

Furthermore, one should bear in mind that any kind of overstress strengthens the material only against loads applied in one direction, but weakens with a load, acting in the opposite direction. Hence, this technique is applicable only for loads acting in one direction, pulsating and alternate, provided the load of one sign is prevailing (asymmetric cycles).

Clearly, any system subjected to the loads of permanent direction and made from sufficiently plastic material, possesses the self-

strengthening property. Temporary increase of the working load up to a value, inducing moderate plastic deformations, strengthens the material. If, however, a part is subjected to alternate loads surpassing the yield point under the unidirectional load action, the material's ability to withstand the load action of opposite sign falls.

The advantage of the overloading method is that it allows the most heavily stressed areas to be relatively strengthened. It seems as if the overload finds the weakest spots in the structure and automatically strengthens them.

(*b*) *Strengthening by Work Hardening*

Another form of plastic strengthening is surface work hardening. It consists in compacting the surface layer to the depth of 0.2-0.8 mm and the induction in this layer of compressive stresses favourable to strength.

Mechanism of the surface hardening is illustrated in Fig. 300*a*. During hardening process the surface layer spreads. If it were able

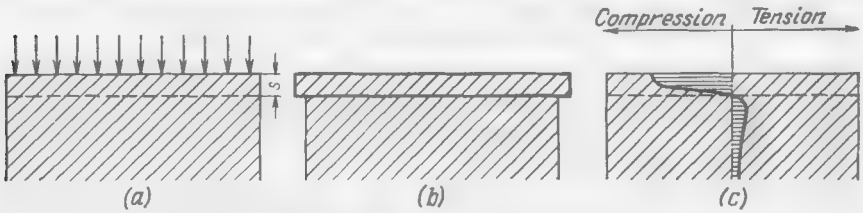


Fig. 300. Effect of work hardening

to extend freely it would separate from the basic metal (Fig. 300*b*), but the extension is prevented by cohesive forces in the metal. As a result, biaxial (longitudinal and transverse) compressive stresses occur in the hardened layer while in the thick basic metal negligible reactive compressive stresses arise (Fig. 300*c*).

In addition the hardening process strengthens the surface layer owing to structural and phase changes occurring in the material. In a beam, bent in a constant direction by a transverse force (Fig. 301), it is advantageous to harden the surface, opposing the active force. The extension of surface layer caused by the hardening is accompanied by bending of the beam in the same direction as the working load. The elastic counteraction of the basic metal tending to unbend the beam will compress the plastically stretched out layers, thus inducing compressive stresses in them (Fig. 301*a*). When applying the working load (Fig. 301*b*), the compressive stresses are subtracted from and lower the tensile stresses (Fig. 301*c*).

With two-sided hardening the picture is somewhat changed. In this case the residual compressive stresses originate from both sides

of the beam (Fig. 301*d*). When adding the residual and working stresses (Fig. 301*e*) the final tensile stresses are decreased and those of compression increased (Fig. 301*f*). But since it is the tensile stress which defines the strength, the resultant load capacity of the part is enhanced. In addition the part acquires the ability to carry higher loads in both directions.

The surface hardening techniques were described in Section 5.

Hardening in a stressed state is extremely effective. This is actually a combination of overloading and hardening. With this method the part is loaded with a force that acts in the same direction as the working load thus producing in the material elastic or plastic deformation. The surface of the part in this condition is subjected to work

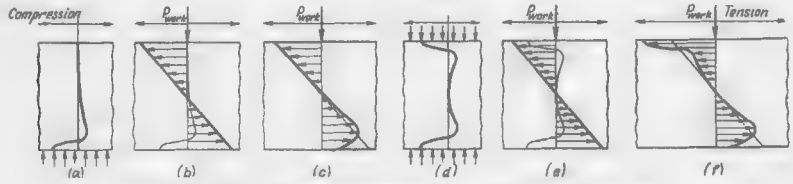


Fig. 301. Strengthening by work hardening

hardening (e.g., shot blast treatment). After removing the load, residual compressive stresses occur in the surface layer, which are much higher than those when overloading or work hardening is performed separately.

Applied recently is an explosive hardening technique. This technique is much superior to others in capacity and versatilities. Explosive hardening enables parts having intricate configurations to be strengthened with simultaneous consolidation of all external and internal surfaces. The consolidation intensity and depth are controlled by the explosion power.

Surface hardening (induction hardening, also hardening of steels which have limited hardenability) and chemical-thermal treatment (case hardening, nitriding) not only strengthen materials but also induce (as work hardening) residual compressive stresses in the surface layer owing to the formation of higher specific volume structures. Extension of the surface layer is impeded by the core which preserves the original pearlite structure. As a result, biaxial compressive stresses occur in the surface layer (in cylindrical parts—triaxial) and in the core only insignificant tensile reactive stresses develop.

(*c*) *Volumetric Consolidation*

Volumetric consolidation is the deep reduction of parts of components undergoing tensile stresses during operation. Generally the reduction takes place when the part is a blank in its cold or semiplastic state (thermal deformation).

Let us now consider a beam being bent by a transverse force P_{work} (Fig. 302). The reduced areas are those opposite to the acting load (hatched section in Fig. 302*a*). Plastic deformation of the material causes the beam to sag. After reduction the elasticity of the material

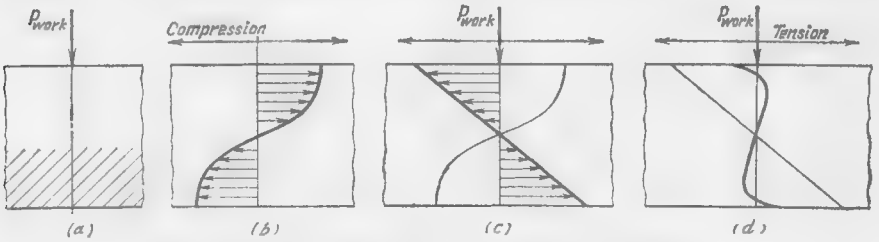


Fig. 302. Strengthening by spatial deformation

straightens the beam. In the reduced areas biaxial compressive stresses arise and in the non-reduced areas tensile stresses occur (Fig. 302*b*). During action of the working load (Fig. 302*c*), the summed residual and working stresses lessen the resultant stresses (Fig. 302*d*).

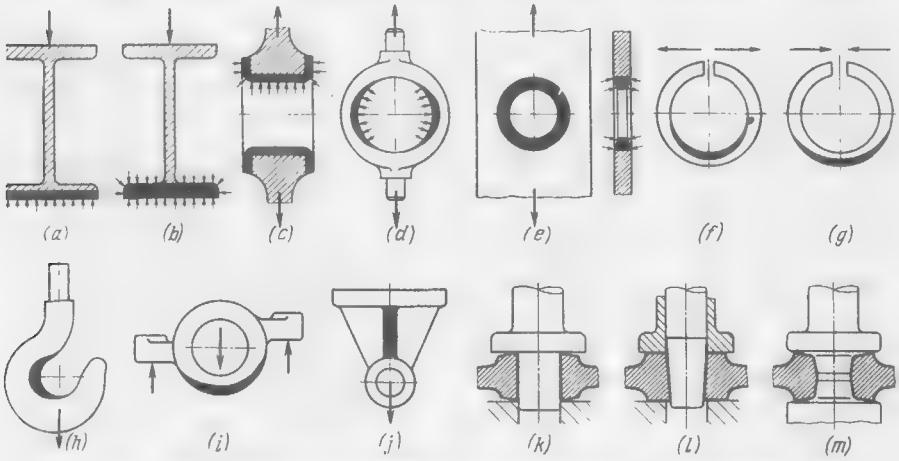


Fig. 303. Examples of strengthening by spatial deformation

The magnitude and distribution of the resultant stresses will depend on the cross-sectional relationship of the reduced and non-reduced zones, on the degree of reduction and the changes along the cross-section of the part. With a rational choice of these parameters it is possible to significantly (sometimes completely) decrease the resultant stresses.

Examples of volumetric consolidation by reduction are illustrated in Fig. 303 (reduced areas are shown black). The beams (Fig. 303*a*, *b*)

are strengthened by rolling the flanges and holes (Fig. 303*c*)—by broaching; flat parts (Fig. 303*e*) by pressing from end faces and rings (Fig. 303*d*, *f*, *g*)—by eccentric burnishing.

Figure 303*k-m* depicts some methods of strengthening of disk-shaped components by metered deformation between plates with a simultaneous enlargement of the hole.

(*d*) *Thermal Strengthening*

The thermal strengthening technique is based on the fact that in the course of uneven heating the hot areas of the part have compressive stresses and the cold areas tensile stresses. The magnitudes of the stresses depend on the temperature gradient, coefficient of

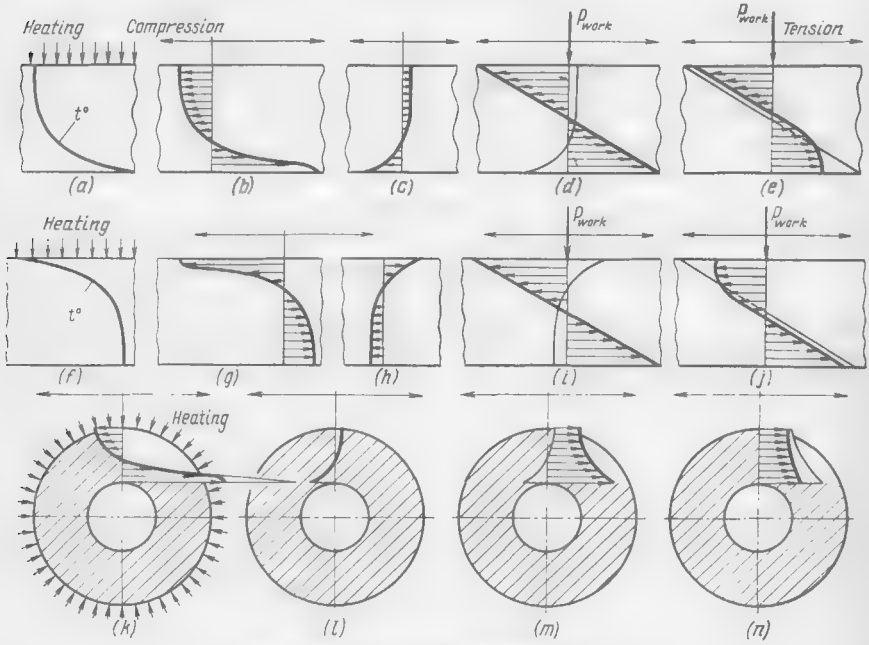


Fig. 304. Thermal strengthening

linear expansion and the elasticity modulus of the material. With sufficiently high temperature gradients local plastic strains arise, which can be utilized for the strengthening.

Let a beam be bent by a working force P_{work} (Fig. 304*a-e*). With thermal strengthening the beam must be heated from the side of acting forces. The heated layers are lengthened and then compressed by adjacent more cold layers, in which tensile reactive stresses appear. The magnitudes of tensile and compressive stresses, as well as their

distribution across the section depend on the temperature gradient in the section. In the case being considered it is advantageous to heat the beam uniformly to a substantial depth (Fig. 304*a*) so that slight compressive stresses appear on the heated side and high, exceeding the yield limit, tensile stresses in the thin cold layer on the opposite side (Fig. 304*b*).

After cooling to the initial temperature, the stresses, which appeared as a result of the temperature difference, disappear and the plastically stretched layers will be compressed by the elasticity of the basic material. As a result, compressive stresses arise in these layers and on the opposite equalizing tensile stresses (Fig. 304*c*). The beam becomes advantageously prestressed. Under the influence of working loads (Fig. 304*d*), the residual stresses are subtracted from the working ones thus reducing the resultant stresses (Fig. 304*e*).

Intensive heating of a thin upper layer (Fig. 304*f*) induces residual compressive stresses in it (Fig. 304*g*); after cooling tensile stresses occur in this layer and in the lower lying layers insignificantly equalizing compressive stresses (Fig. 304*h*). When applying the working force P_{work} the residual stresses are subtracted from the working ones (Fig. 304*i*), thus the resultant stresses (Fig. 304*j*) reduce. It should be noted, however, that the gain in the value of the tensile stresses is much lower than in the previous case.

Thermal strengthening is most often applied to engineering parts made from light alloys, which possess a number of the necessary properties for the case in question, i.e., high coefficient of linear expansion, low yield point and low transition temperature to the plastic state. Thermal strengthening is applied to light alloy high-speed rotors. The problem in this case is to balance centrifugal tensile stresses which are maximum at the rotor hub. Still higher tensile stresses arise in the hub if rotor during work heats from the periphery and still more if the hub is press-fitted upon the shaft.

Figure 304*k-n*, shows thermal strengthening of a disk, subjected to tensile working stresses, whose curves are presented in the form of epure in Fig. 304*m*. The heating of the disk begins from its periphery (Fig. 304*k*). The heating temperature and the temperature gradient along the radius are selected so that tensile residual stresses are induced in the internal cold layer. After cooling, the stretched layers are compressed by the elasticity of the external layers. In the internal layers compressive prestresses are produced and tensile prestresses in the external layers (Fig. 304*l*). When the working load is applied (Fig. 304*m*), the residual and working stresses add algebraically; the resulting stresses (Fig. 304*n*) are less and more favourably distributed than in a disk not thermally strengthened.

The actual thermal strengthening procedure may vary. Temperature gradient can be increased by simultaneously cooling the hub and heating the periphery. Sometimes deep cooling of the hub will suf-

fice (e.g., cooling the hub with liquid air). The procedure sequence may also be different. It is possible to uniformly heat the disk and then quickly cool the hub. The same result is obtained when the disk is first cooled to a minus temperature and then heated from its periphery.

Thermal strengthening must agree with the magnitude and sign of working stresses. If the core of the part is subjected to compressive stresses (e.g., to compressive thermal stresses, arising when the working temperature of internal layers is higher than that of the external ones), then the aim of the thermal strengthening procedure is to obtain tensile prestresses in the internal layers by prematurely providing residual compressive strains in these layers. In this case the process of thermal strengthening must be opposite to the above, i.e., the part has to be heated from inside and cooled on the outside.

To generalise the situation: compressive prestresses require cooling of the part area; tensile prestresses—heating.

The drawback of the thermal hardening method is the delivery of factors which define the magnitude, sign and distribution of prestresses, and also the difficulty of strictly maintaining thermal conditions which predetermine the stability and reproducibility of the results.

The thermal strengthening temperature should be lower than that of the previous heat treatment, otherwise the thermal strengthening effect is lost.

If the part, when working, heats to a temperature close to the strengthening temperature, especially in the presence of high stresses, the strengthening effect will again be lost.

(*e*) *Improving Strength of Trussed Systems*

Similar methods are used to improve the strength of trussed and analogous structures.

As an example let us consider the case of a trussed bracket, loaded with tensile force P_{work} (Fig. 305*a*). The bracket middle bar carries more load than the side bars. The elastic deformation of the middle bar under the load (and, hence, according to Hooke's law—the tensile stresses in the bar) exceeds the deformations of the side bars in the proportion $s/s' \approx 1/\cos \alpha$ (when $\alpha = 60\text{-}70°$, $s/s' = 2\text{-}3$).

The system may be strengthened by overloading, causing tensile plastic deformations in the middle bar. After removing the strengthening load, the middle bar will be compressed due to the elasticity of the side bars (Fig. 305*b*) in which arise tensile stresses (light arrows). If such a prestressed system is subjected to the working load then the addition of the residual and working stresses, i.e., the load on the bars becomes more uniform (Fig. 305*c*) and the load capacity of the system is enhanced.

Load capacity can also be increased by thermal strengthening, i.e., by heating the side bars until residual tensile strains are produced in the middle bar. After cooling, in the middle bar there occur compressive stresses and the system becomes prestressed.

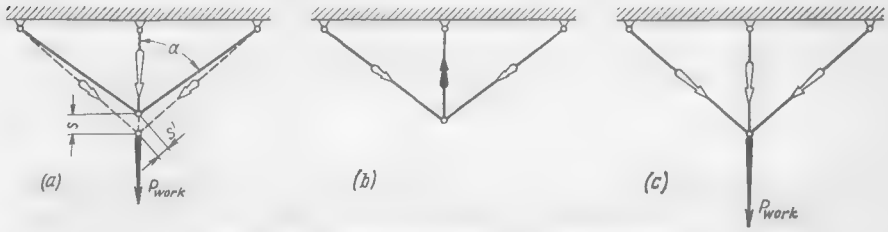


Fig. 305. Strengthening of a truss system

It is possible to employ an elastic strengthening method. For this the side bars should be pulled tight or the length of the middle bar increased above the nominal so that compressive stresses are induced in it during assembly.

In castings beneficial prestresses can be assured by earlier crystallization of material of the middle bar and by cooling it quicker in comparison to the side bars (e.g., by providing cooling condensers in the mold).

Surface finish

Any machined part will have on its surface some minute irregularities, whose size, shape and location will depend on the kind of machining and the type of cutting tool used (Fig. 306). Such super-

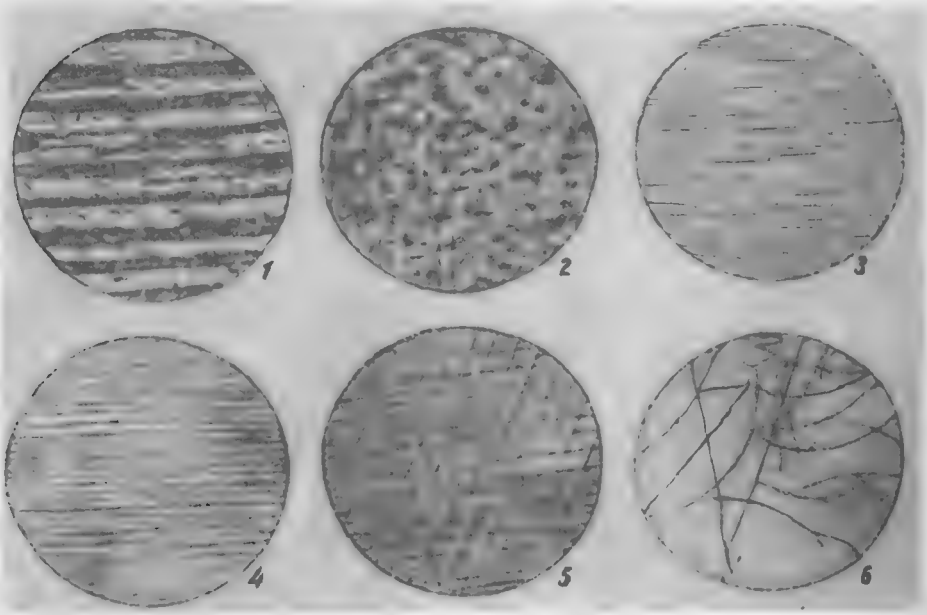


Fig. 306. Microphotographs of surfaces machined by different methods (X30)
1—rough turning; *2* — reaming; *3* — diamond boring; *4* — honing; *5* — fine honing; *6* — superfinishing

ficial irregularities appreciably affect the surface behaviour and service properties of the part.

Rough machining (e.g., rough turning) leaves on the surface notches up to 0.3 mm deep and also ridges with rugged edges. Plastic shear and tearing of metal particles by the cutting tools causes high

residual tensile stresses in the surface layer. The flakes and microcracks which deteriorate the strength of the surface layer become sources of local corrosion. Furthermore, as the metal heats during machining structural changes occur in the surface layer (phase thansformations, recrystallization, etc.), which change the mechanical properties of the metal.

Finishing operations are aimed at the complete or partial elimination of the damaged and overstressed layer, removing or smoothing out the irregularities.

Fig. 307. Microprofiles

As a rule, improvements in smoothness of surface better fatigue strength, improve wear- and corrosion-resistance of parts, help to maintain dimensional stability of parts during operation, increase contact rigidity, assure uniform load distribution and reduce the friction factor in movable joints.

Generally the quality of a surface (or surface finish) is evaluated through the averaged values of microirregularities, existing on it, in other words, the degree of roughness, though traditionally the term "surface finish" (opposite in meaning to the above) is used.

Evaluating the surface finish by the averaged height of microirregularities is incomplete as it does not account for the nature of microrelief, i.e., the distribution density of the irregularities, their profile, sharpness of undercuts at valley bottoms and other factors which determine wear-resistance, fatigue strength and contact rigidity of the surface.

The microrelief may consist of an alternating series of ridges and valleys (Fig. 307*a*), which may be rarely spaced (Fig. 307*b*), or ridges separated by plateaus (Fig. 307*c*), or valleys separated by plateaus (Fig. 307*d*). The parameters of surfaces in the above mentioned cases would be different although in the case of equal mean height of the microirregularities the quality of surface is accepted as homogenious.

Being only a geometrical index, the averaged height of microirregularities does not reflect the physical and mechanical changes, which occur in the surface layer in the machining process. Nevertheless, it provides, in general, true picture of the quality of surface, because damages, as a rule, are the lesser, the finer the machining.

9.1. Classes of Surface Finish

The Soviet State Standard GOST 2789-59 stipulates the following rules for determining the roughness of a surface.

The base for determining magnitudes of the microirregularities is the mean profile line $m—m$ (Fig. 308*a*), which runs parallel to the geometrical surface of the profile, dividing it such that within the

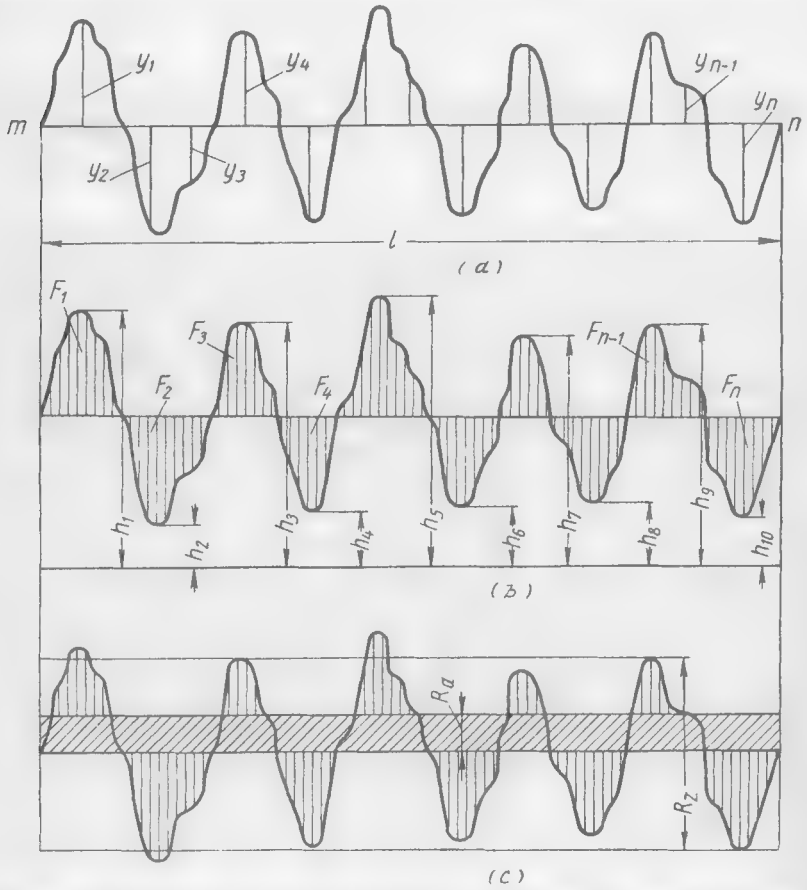


Fig. 308. Determination of mean profile line, arithmetic mean deviation R_a and height R_z of irregularities

limits of the measured part (base length l) the sum of the squares of the heights y_1, y_2, y_3, . . ., y_n, taken at the closest distances one from another, is minimum.

The mean line of the profile may be approximately considered as the line which divides within limits of the base length so that the

areas on both sides of this line are equal (Fig. 308*b*)

$$F_1+F_3+\ldots+F_n=F_2+F_4+\ldots+F_{n-1}$$

Surface roughness is determined from the following parameters.

Mean arithmetic deviation R_a is the mean value of the heights (y_1, y_2, y_3, . . ., y_n) of points on the measured profile (Fig. 308*a*) from the mean line (regardless of algebraic sign)

$$R_a=\frac{1}{l}\int_a^l [y]\,dx$$

Approximately

$$R_a=\frac{1}{n}\sum_1^n [y]$$

Height of irregularities R_z is the mean distance between five highest crests and five lowest valleys within base length l (Fig. 308*b*) measured from a line, running parallel to the mean line

$$R_z=\frac{(h_1+h_3+\ldots+h_9)-(h_2+h_4+\ldots+h_{10})}{5}$$

The value of R_a can be represented as the height of a rectangle whose area is equivalent to the area of profile on each side of the mean line, while the R_z value can be expressed as the mean height of irregularities at the extreme points of the profile (Fig. 308*c*) R_z is considerably (4-5 times) higher than R_a. In practice the R_z value expresses the roughness level more vividly than the R_a value.

GOST 2789-59 establishes 14 classes of surface roughness (Table 28). The classes of surface roughness from 6 to 14 are subdivi-

Table 28

Classes of Surface Roughness

Class	R_a, μm (not more than)	R_z, μm (not more than)	Base length l, mm	Class	R_a, μm (not more than)	R_z, μm (not more than)	Base length l, mm
1	80	320	8	9	0.32	1.6	0.25
2	40	160		10	0.16	0.8	
3	20	80		11	0.08	0.4	
				12	0.04	0.2	
4	10	40	2.5				
5	5	20		13	0.02	0.1	0.08
				14	0.01	0.05	
6	2.5	10	0.8				
7	1.25	6.3					
8	0.63	3.2					

ded into subclasses *a*, *b* and *c*, which allow finer classification within each of the classes. The R_a scale is regarded as basic for classes from 6 to 12 and the R_z scale—for classes from 1 to 5 and from 13 to 14 (thick black lines in Table 28). The R_a and R_z values are correlated to each other as folows: $R_z = 4R_a$ for classes 1-6; $R_z = 5R_a$ for classes 7-14.

In drawings the surface roughness classes are denoted by an equilateral triangle ▽ with a number, indicating the class (and accompanied by a letter of the subclass whenever necessary). The numerical values of the microirregularities, which underlie the classification into classes, only limit their maximum heights. Should the maximum and minimum values of microirregularities be necessary, then two class numbers are quoted. For instance, writing V9-V10 indicates that the roughness must be within the R_a and R_z values, for classes 9 and 10.

Surfaces rougher than the 1st class are denoted by the symbol ∨, above which the ultimate height of irregularities R_z is given in microns. For example, the $\overset{500}{\vee}$ symbol indicates that the acceptable height of surface irregularities shall not exceed 500 μm.

Surfaces, whose roughness does not require any classification, are shown by the symbol ∾.

One should identify certain contradiction, which exists between what is implied by the surface finish designation and what is implied by the symbols depicted on drawings. In accordance with GOST 2789-59 the class of surface finish indicates merely the degree of surface roughness and ignores the method by which this surface has been produced—by machining or in as-received state (e.g., moulding, casting, etc.). Clearly, this is insufficient because shop production requirements reject such ambiguity and suggest that all the surfaces to be machined are strictly specified.

For this reason the appropriate symbols of surface finish are depicted on the drawings only against those surfaces that require definite machining. The rest of the surfaces are denoted by the symbol ∾, regardless of actual surface finish of these surfaces which, in fact, remains in the as-received state.

Thus, the symbol ▽ with its accompanying figure shows that the surface so designated must be machined to the denoted standard of surface finish, while the ∾ symbol indicates the black surface in as-received state.

The values of R_a and R_z parameters are presented on a log scale in Fig. 309, for the different classes and subclasses of surface roughness.

Table 29 cites the classes of surface finish obtainable by various machining operations. Surface roughness classes mostly used in mechanical engineering, are listed in Table 30 for reference.

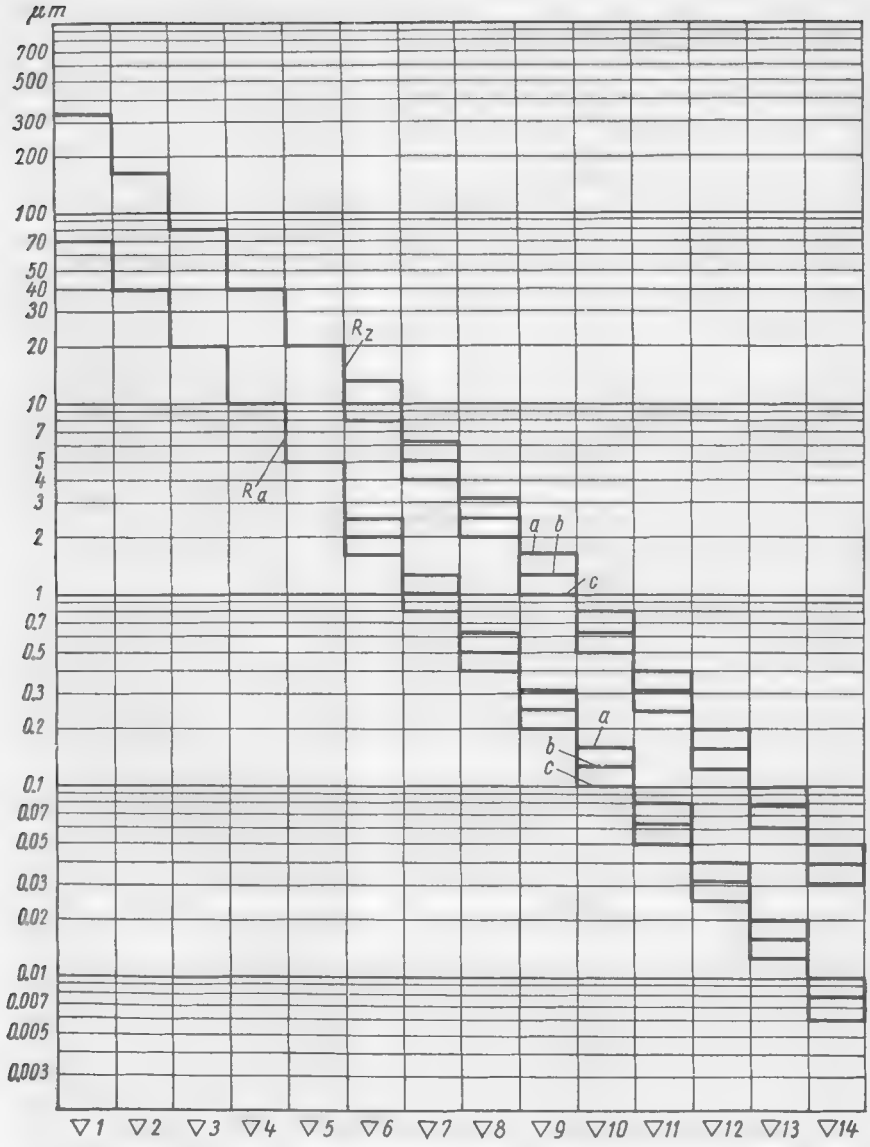


Fig. 309 R_a **and** R_z **values for various classes of surface finish**

Surface Roughness Values Obtainable

Class of surface roughness		▽ 1	▽ 2	▽ 3	▽ 4
R_z, μm, not more than		320	160	80	40
R_a, μm, not more than		80	40	20	10
Gas cutting (machine) Filing Drilling		—	—	—	
Planing	finish finest				—
Face milling	finish finest				
Circular milling	finish finest				
Turning	finish finest				
Boring	finish finest				
Core drilling					
Face undercutting	finish finest				
Cutting of male thread	tool, chaser, circular die, screw die, milling rolling grinding				
Cutting of female thread	tap, tool milling grinding				
Machining of gears	shaping, milling hobbing shaving				

Table 29

by Various Machining Methods

▽ 5	▽ 6	▽ 7	▽ 8	▽ 9	▽ 10	▽ 11	▽ 12	▽ 13	▽ 14
20	10	6.3	3.2	1.6	0.8	0.4	0.2	0.1	0.05
5	2.5	1.25	0.63	0.32	0.16	0.08	0.04	0.02	0.01

Class of surface roughness		▽ 1	▽ 2	▽ 3	▽ 4
R_z, μm, not more than		320	160	80	40
R_a, μm, not more than		80	40	20	10
Machining of gears	grinding finishing, generating				
Anode-mechanical machining	ordinary fine				
Electro-chemical machining to size	ordinary fine				
Electric discharge machining	usual fine				
Ultrasonic machining (holes, recesses)					
Scraping	usual fine				
Reaming	usual fine				
Broaching	usual fine				
Plane grinding	usual fine				
Circular grinding	usual fine				
Regrinding	usual fine				
Polishing	usual fine				

Table 29 (continued)

▽ 5	▽ 6	▽ 7	▽ 8	▽ 9	▽ 10	▽ 11	▽ 12	▽ 13	▽ 14
20	10	6.3	3.2	1.6	0.8	0.4	0.2	0.1	0.05
5	2.5	1.25	0.63	0.32	0.16	0.08	0.04	0.02	0.01

Class of surfaces roughness		▽ 1	▽ 2	▽ 3	▽ 4
R_z, μm, not more than		320	160	80	40
R_a, μm, not more than		80	40	20	10
Honing	usual fine				
Superfinishing	finest repeated				
Ball burnishing (sizing holes)		1-2 classes higher than			
Rolling		1-2 classes higher than			
Shot-blasting		1-2 classes lower than			
Liquid polishing (hydrohoning)		2-3 classes higher than			
Electropolishing		2-3 classes higher than			

Table 29 (continued)

▽ 5	▽ 6	▽ 7	▽ 8	▽ 9	▽ 10	▽ 11	▽ 12	▽ 13	▽ 14	
20	10	6.3	3.2	1.6	0.8	0.4	0.2	0.1	0.05	
5	2.5	1.25	0.63	0.32	0.16	0.08	0.04	0.02	0.01	
										the original surface finish (up to class 10)
										the original surface finish (up to class 11)
										the original surface finish
										the original surface finish (up to class 12)
										the original surface finish (up to class 14)

Table 30

Surface Finish of Finished Blanks

Class of surface finish		▽ 1	▽ 2	▽ 3	▽ 4	▽ 5	▽ 6	▽ 7	▽ 8	▽ 9	▽ 10
R_z, μm, not more than		320	160	80	40	20	10	6.3	3.2	1.6	0.8
Rolled metal sections		—	—	—	—						
Casting	floor mould	—	—	—							
	core mould		—	—	—						
	shell mould			—	—	—					
	metal mould				—	—	—				
	investment				—	—	—	—			
	pressure (non-ferrous alloys)					—	—	—	—		
Hot stamping	ordinary precision			—	—	—	—	—			
Precision forging (coining)							—	—	—	—	
Drawing								—	—	—	
Plastic products (pressing and press-moulding)									—	—	—

9.2. Selection of Surface Finish Classes

The class of surface finish must be agreed with the class of manufacturing accuracy. The higher the accuracy class, the higher must be the surface finish. Otherwise, the value of the surface microirregularities becomes commensurable with tolerance margins. Measurements taken over the extreme points of the profile give false dimensional values. In use the part quickly loses its accurate dimensions due to wear and crushing of ridges (in movable joints) or the action of working loads (in fixed joints).

For classes of rougher manufacturing accuracy with wider tolerances the class of surface finish can be lowered, thus reducing production costs.

To obtain different classes of accuracy the minimum surface finish class is as follows:

Class of accuracy	1	2	2a	3	3a	4	5	7
Class of surface finish	▽7	▽7	▽6	▽5	▽5	▽4	▽4	▽4

When selecting the surface finish class properties of the material and hardness of the part surface must be considered. For steels a high degree of surface finish can be obtained with hardness values not below 30-35 Rc. Steel products, undergoing finishing treatment, must be at least structurally improved or normalized. Fine finishes on raw low-carbon steels are difficult to obtain.

Due to machining conditions holes are more difficult to finish than shafts. For this reason the surface finish standards assigned to holes should be somewhat lower than those for shafts.

In general lower classes of surface finish should always be accepted if commensurate with reliable component operation, since higher surface finish requirements mean additional finishing operations and, hence, increased production costs.

Furthermore, higher surface finish does not always better the functioning of joints. Press-fitted joints, for example, have certain optimum standards of surface finish and deviations from these standards to any side worsen joint strength.

In the interest of economy free surfaces (i.e., those, which are either out of engagement or are separated by a gap from neighbouring surfaces) should be machined to the lowest possible surface finish standard. Exceptions are heavily-loaded parts subjected to cyclic loads. To attain higher fatigue strength such parts are machined, polished or burnished all over to high surface finish standards.

Listed below and taken on the basis of general engineering practice are approximate values of surface finish for typical engineering parts.

Surface Finish of Typical Engineering Parts

Plain bearings:	
low-loaded, running at moderate surface speeds:	
hole	▽7—▽9
shaft	▽8—▽10
high-loaded, running at high surface speeds:	
hole	▽8—▽10
shaft	▽10—▽12
Plain thrust bearings (working surfaces):	
low-loaded	▽6—▽8
high-loaded, running at high surface speeds	▽8—▽12
Spherical surfaces (of self-aligning bearings, etc.)	▽9—▽12
Fixed connections accomplished with a sliding fit:	
hole	▽8—▽9
shaft	▽9—▽11
Connections with transition fits:	
hole	▽7—▽9
shaft	▽8—▽10

Pressed connections:	
hole	∇7—∇10
shaft	∇8—∇11
Thrust shoulders of fixed cylindrical connections (working surfaces)	∇6—∇8
Fits of antifriction bearings:	
hole in housing, for bearings of class:	
standard (H)	∇8—∇9
higher (П)	∇9—∇10
high (B)	∇10—∇11
precision (A)	∇11—∇12
shaft, for bearings of class:	
standard (H)	∇8—∇10
higher (П)	∇10—∇11
high (B)	∇11—∇12
precision (A)	∇12—∇13
Rolling contact bearings, contact load connections	∇10—∇13
Cylinders (highly polished):	
for pistons with soft packing (cups)	∇7—∇10
for pistons with metallic rings	∇9—∇12 (lapped in)
Pistons (working surfaces):	
cast iron and steel	∇9—∇10
light alloy	∇10—∇12
Piston gudgeon pins:	
hole	∇8—∇11
pin	∇9—∇12
High-pressure plunger pumps:	
cylinders	∇10—∇12 (lapped in)
plungers	∇12—∇14
Cylindrical slide valves:	(lapped in)
oil distributing:	
hole	∇7—∇9 (lapped in)
valve	∇8—∇11 (lapped in)
gas distributing:	
hole	∇9—∇11 (lapped in)
valve	∇9—∇12 (lapped in)
Flat slide valves:	
body	∇7—∇10 (lapped in)
gate	∇8—∇11 (lapped in)
Taper plug cocks (working surfaces):	
hole	∇8—∇10 (lapped in)
plug	∇10—∇12 (lapped in)
Valves with conical sealing surfaces:	
guide surfaces:	
guide	∇8—∇9
stem	∇9—∇10

sealing surfaces:	
seat working surface	∇9—∇11 (lapped in)
valve working chamfer	∇10—∇12 (lapped in)
Cam mechanisms (working surfaces):	
cam	∇9—∇11
drive roller	∇9—∇12
tappet	∇8—∇11
Templets (working surfaces):	
templet	∇8—∇10
roller	∇9—∇11
Splined connections (centering surfaces):	
centering on outer diameter:	
hole	∇7—∇10
shaft	∇8—∇10
centering on inner diameter:	
hole	∇9—∇12
shaft	∇7—∇9
centering on spline faces:	
female surfaces	∇7—∇10
male surfaces	∇8—∇11
splined connections with clearance:	
spline (working faces)	∇7—∇10
hole	∇8—∇11
shaft	∇7—∇8
Keyed connections (working faces):	
keyways	∇5—∇7
keys	∇6—∇8
Vee ways:	
female surfaces	∇8—∇10
male surfaces	∇9—∇12
Male threads:	
commercial grade	∇5—∇6
higher accuracy	∇6—∇7
precision	∇7—∇9
Female threads:	
commercial grade	∇4—∇5
higher accuracy	∇5—∇6
precision	∇6—∇8
Leadscrews (working surfaces):	
nut	∇8—∇10
screw	∇8—∇12
Spur gears (working faces teeth):	
non-critical	∇6—∇7
operating under moderate loads and surface speeds	∇7—∇8
operating under medium loads and surface speeds	∇9—∇10
heavily-loaded, subjected to impact loads and running at high surface speeds	∇10—∇12 (lapped in or run-in)
Helical and herringbone gears (working faces of teeth):	
operating under moderate loads and moderate surface speeds	∇6—∇8
heavily-loaded and running at high surface speeds	∇8—∇10

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